RL-TR-96-159
Final Technical Report
May 1997

PAT JITTER STABILIZATION

CALSPAN - UB RESEARCH CENTER

Victor A. Skormin

19971007 214

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Rome Laboratory
Air Force Materiel Command
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19971007 214

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APPROVED: D J Nicholson

DONALD J. NICHOLSON, Dr.
Project Engineer

FOR THE COMMANDER: John A Graniero

JOHN A. GRANIERO
Chief Scientist
Command, Control & Communications Directorate

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REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (Leave Blank)	2. REPORT DATE May 1997	3. REPORT TYPE AND DATES COVERED Final Dec 94 - Dec 95

4. TITLE AND SUBTITLE PAT JITTER STABILIZATION	5. FUNDING NUMBERS C - F30602-93-D-0075/T23 PE - 62702F PR - 4519 TA - 63 WU - P7
6. AUTHOR(S) Victor A. Skormin	
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) CALSPAN - UB RESEARCH CENTER P.O. Box 400 Buffalo, NY 14225	8. PERFORMING ORGANIZATION REPORT NUMBER N/A
9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES) Rome Laboratory (C3BA) 525 Brooks Road Rome, NY 13441-4505	10. SPONSORING/MONITORING AGENCY REPORT NUMBER RL-TR-96-159

11. SUPPLEMENTARY NOTES

Rome Laboratory Project Engineer: Donald J. Nicholson /C3BA/(315) 330-7437

12a. DISTRIBUTION/AVAILABILITY STATEMENT Approved for Public Release: Distribution Unlimited.	12b. DISTRIBUTION CODE

13. ABSTRACT (Maximum 200 words)

This report contains the results of our most recent efforts in providing both theoretical and experimental information on selection of control systems for stabilizing optical beams used as signal carriers in communications systems, with the major system goal being high speed inter-satellite links.
The work described here applies advanced control techniques to improving the dynamic performance of a fast steering mirror in the presence of platform vibration; the response of the mirror was analyzed, revealing the existence of cross-channel coupling resulting from mechanical bending modes. The model reference control approach was then employed in designing an adaptive modification of the existing control system, and the results experimentally verified using the PAT testbed at Rome Laboratory.

14. SUBJECT TERMS Control systems, optical beam steering	15. NUMBER OF PAGES 84
	16. PRICE CODE

17. SECURITY CLASSIFICATION OF REPORT UNCLASSIFIED	18. SECURITY CLASSIFICATION OF THIS PAGE UNCLASSIFIED	19. SECURITY CLASSIFICATION OF ABSTRACT UNCLASSIFIED	20. LIMITATION OF ABSTRACT UL

NSN 7540-01-280-5500

Standard Form 298 (Rev. 2-89)
Prescribed by ANSI Std. Z39-18
298-102

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1. GENERAL

Adaptive feedback and feedforward control techniques can significantly improve accuracy and dynamic performance of the existing pointing-acquisition-tracking (PAT) mechanisms used in laser communication systems [3,4,5]. Although such systems utilize the most advanced materials and technologies, their performance is adversely affected by the exposure to jitter and natural transients (bending modes) of moving mechanical parts. The main motivation for this research is in the fact that improving the accuracy of a laser beam steering system has the potential for replacing an expensive, high performance optical mirror by a lower cost mirror with enhanced performance of control system. In addition, improvement of beam positioning accuracy can result in the reduction of the required power of lasers used as communication medium [5].

The study was aimed at the application of advanced control techniques for further improvement of dynamic performance of a 500Hz fast steering mirror of a pointing-acquisition-tracking (PAT) system for lasercom. First, a series of laboratory tests was conducted leading to the analysis of dynamic characteristics of the mirror. Recorded responses of the azimuth and elevation positions of the mirror were utilized for the development of a detailed mathematical model of the mirror, featuring "main dynamics" of the elevation and azimuth channels, bending modes, and cross coupling between the channels. The model was implemented in software and its parameters were estimated by matching simulated responses to experimental data.

The mathematical and simulation models obtained herein, were used for comprehensive investigation of the mirror performance and analysis of its compliance with design specifications. The mathematical description provided the basis for the modification of control procedures resulting in the extension of performance limits of the mirror hardware. The model-based simulation scheme was used to define and evaluate the reference model, feedback and feedforward controllers, assuring that improvements in mirror

performance are weighted against magnitudes of control efforts.

The model reference (MR) approach of adaptive control was used for the modification of the existing control system of the 500Hz mirror. The reference model, feedback and feedforward controllers were defined in terms of transfer functions and software procedures. Application of the MR approach allowed for the cancellation of undesirable components of mirror dynamics (bending modes). The MR approach provided the basis for the elimination of cross-coupling between azimuth and elevation channels. Frequency response of the particular channels of the mirror was flattened and significantly extended. The study provided the evidence that scope of applications of the 500 Hz mirror could be widened by improving its control system.

Effects of satellite jitter on the beam positioning system are quite similar to the effects of bending modes of the optical mirror. A simulation study of the application of the MR approach for the reduction of jitter effects on a high precision positioning system was conducted.

Preservation of performance characteristics of a PAT system subjected to long term effects of adverse operational conditions and natural wear presents an important problem. A simulation study of the application of the MR approach for improving robustness of a high precision positioning system was conducted.

The experimental part of this study was conducted on the PAT testbed of Rome Laboratory. The computer simulation work utilized software package CC providing an ***analytical solution*** to the system analysis and control problems [9].

2. EXPERIMENTAL ANALYSIS OF A 500Hz OPTICAL MIRROR

Mathematical model of a 500 Hz beam steering mirror, including its control circuitry, is based on the series of transient responses recorded during laboratory testing. The testing procedure implied that step voltage signals were applied to the azimuth (X) and elevation (Y) inputs of the mirror control system. The responses of the azimuth (X) and elevation (Y) positions of the mirror were represented by the output voltage signals of built-in sensors of the mirror system. Step responses of the mirror channels were recorded repeatedly at different magnitudes of the input signals in order to assess the linearity of mirror characteristics and repeatability of results. Generation of the test signals and recording of testing results were performed using a PC equipped with software and interface of LabView package. The scheme shown in Figure 2-1 illustrates the testing setup.

Testing results are given as follows:

Figure 2-2a shows the family of responses of the azimuth position to the .5, 1., 1.5, 2., 3., and 3.2 V signals applied to the azimuth input (X-to-X response). Figure 2-2b shows one of the step responses recorded at a much higher sampling frequency. Additional records of transient responses of the azimuth channel are given in figures A-1, A-2, and A-3 of the Appendix.

Similarly, Figures 2-3a,b shows the family of responses of the elevation position to the 1., 1.5, 2., 2.5, 3., and 3.5 V reference signals (Y-to-Y response). Additional records of transient responses of the elevation channel are given in figures A-4, A-5, and A-6 of the Appendix.

Figures 2-3a,b,c,d,f represent crosscoupling between the azimuth and elevation channels, providing an evidence that crosscoupling can be viewed as the "exchange of bending modes" between channels with some residual effects. Additional records of transient responses of the elevation channel are given in Figures A-7 of the Appendix.

Figure 2-1. Schematics of the testing setup.

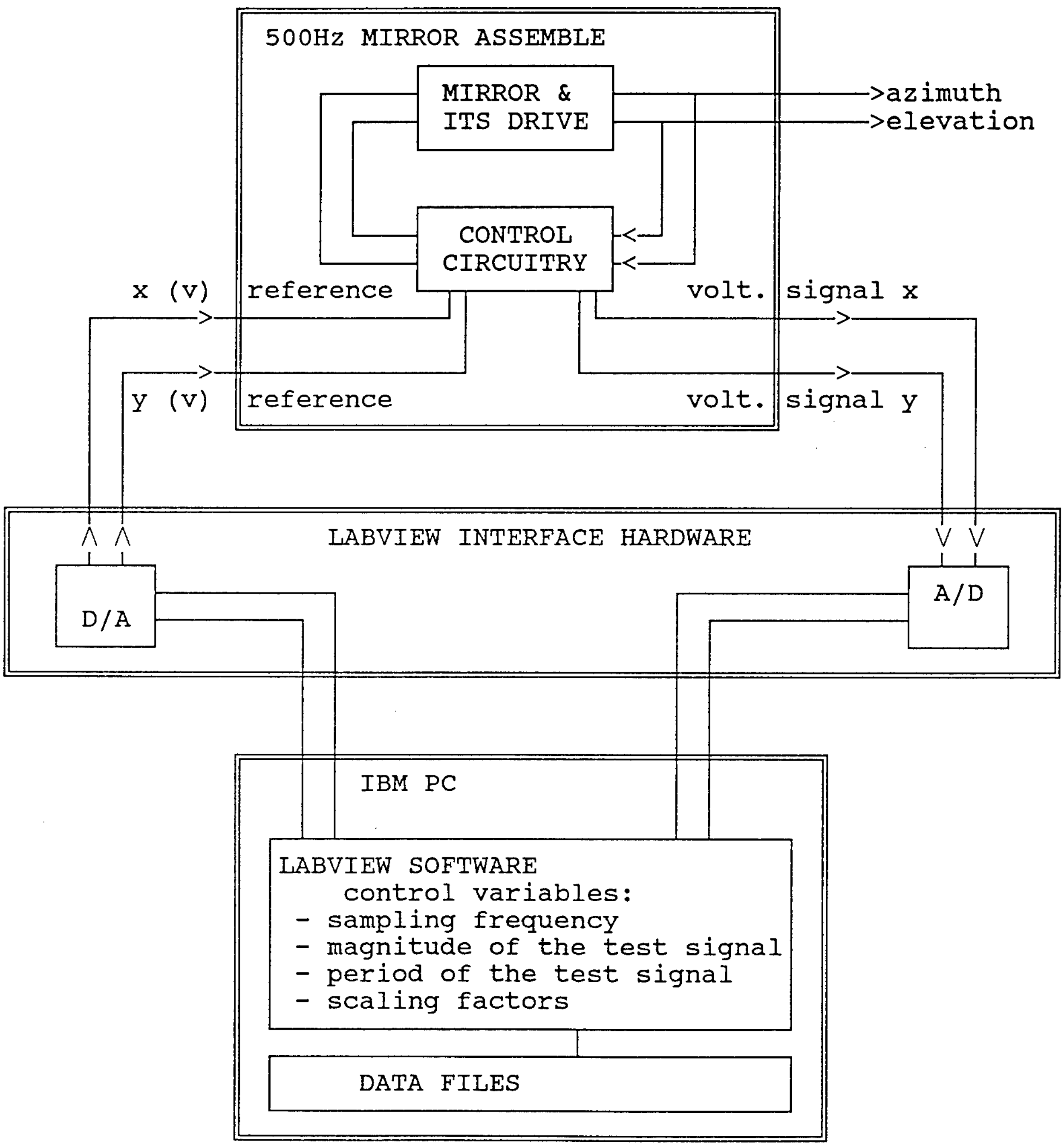


Figure 2-2. Recorded step response of the azimuth channel

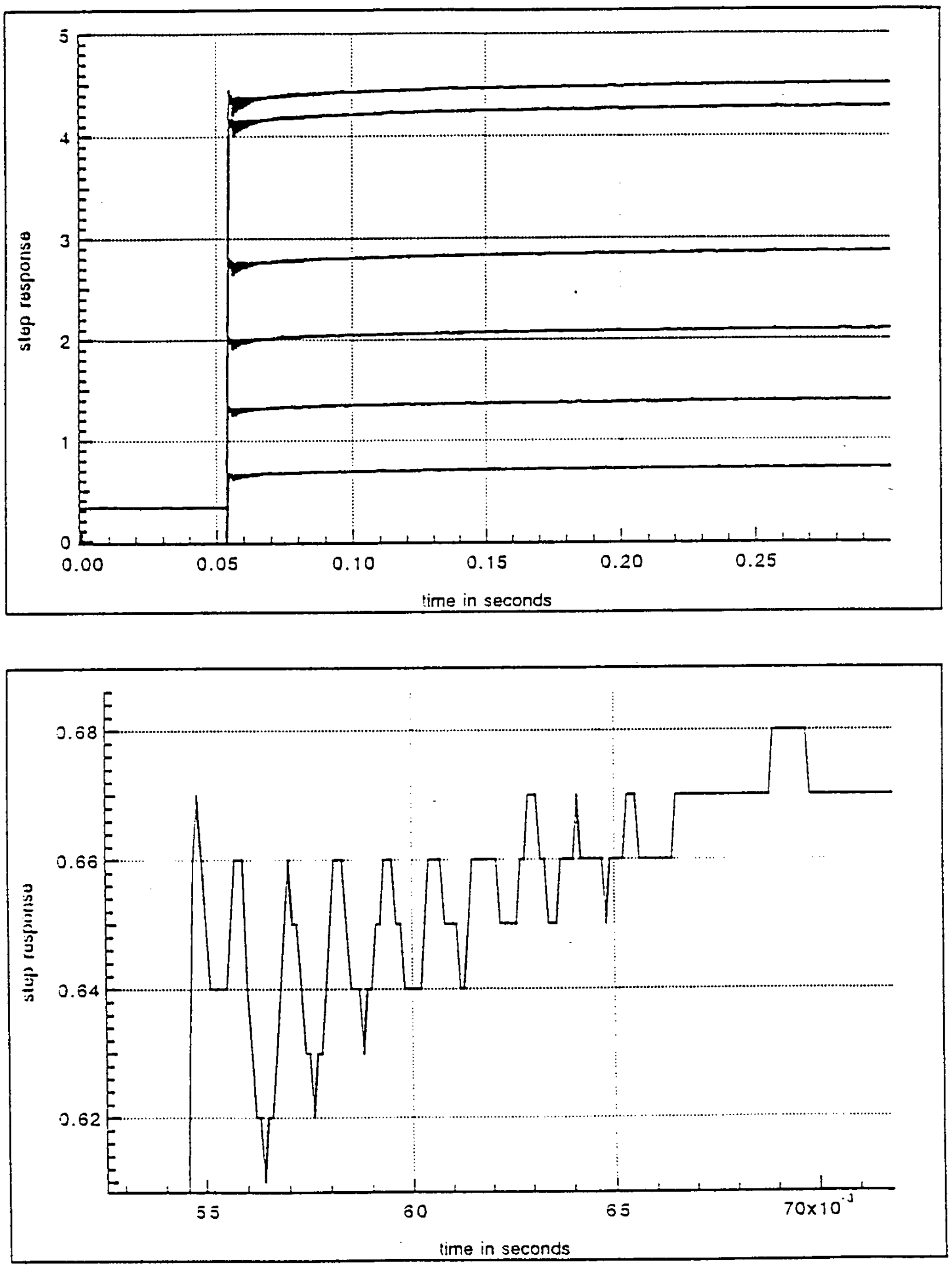


Figure 2-3. Recorded step response of the elevation channel

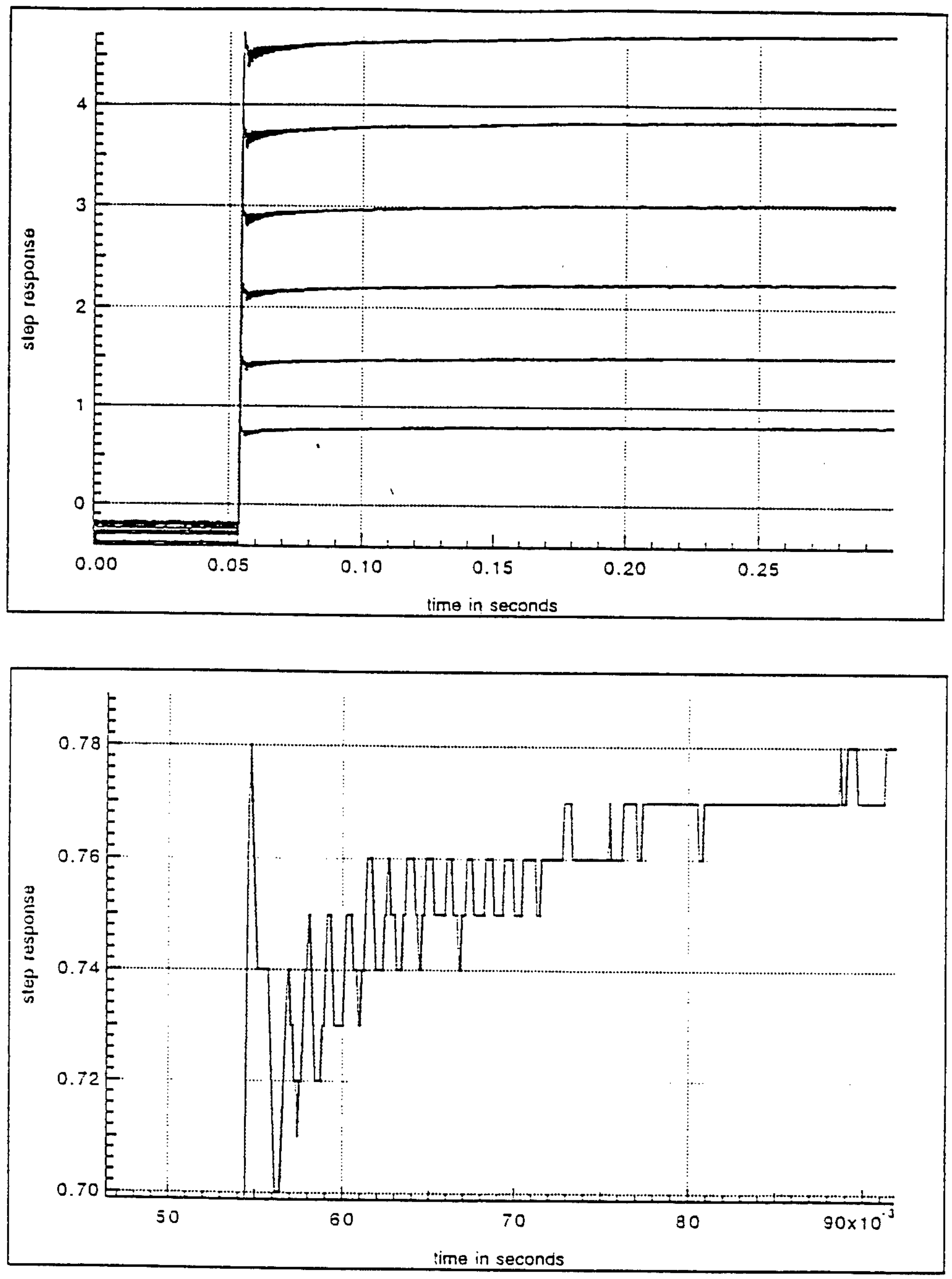
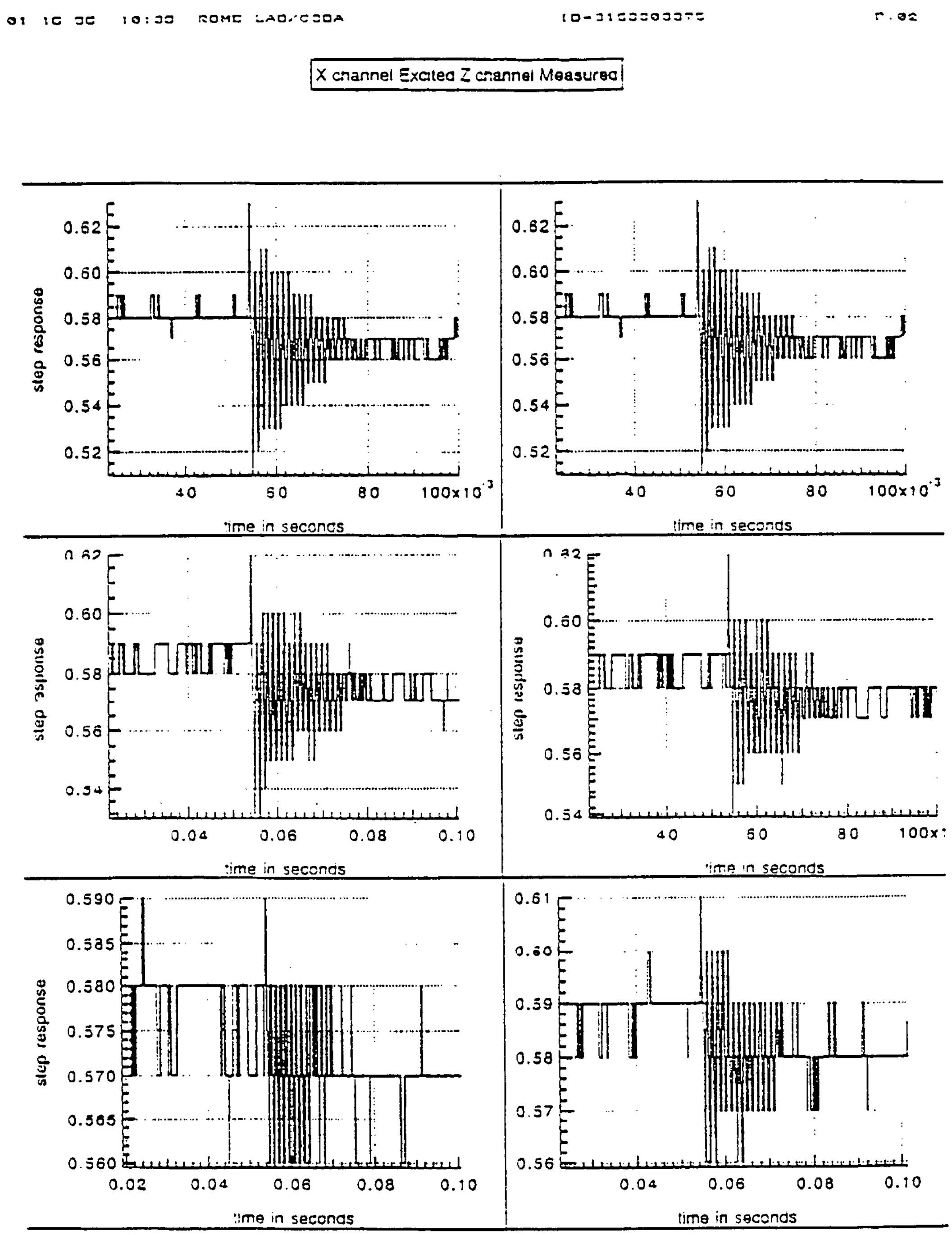


Figure 2-4. Recorded cross-coupling interaction between the elevation and azimuth channels



3. MATHEMATICAL DESCRIPTION AND COMPUTER SIMULATION OF THE 500Hz MIRROR

Recorded step responses of the azimuth and elevation channels of the mirror were sufficient for the evaluation of the transfer functions of the mirror channels. This task was performed by

- interpretation of the recorded responses as a combination of step responses of "typical" dynamic systems,
- parameter estimation of "typical" systems accountable for the frequency, overshoot, and settling time of each component of the recorded response, and
- estimation of "weights" of each component accountable for the overall shape and steady-state value of the recorded response.

The mathematical description task utilized analysis of step response of many "candidate models" provided by CC software, and can be viewed as an "intelligent trail-and-error" process.

Analysis of the obtained responses indicates that X-to-X dynamics exhibits a combination of

- dynamics of a first order system with settling time of .02 seconds,
- dynamics of a second order system with settling time of .0005 seconds, natural frequency of 3800 rad/sec, damping ratio of .5,
- dynamics of a second order system with settling time of .0095 seconds, natural frequency of 3500 rad/sec, damping ratio of .03, and constant gain of 0, and
- dynamics of an integrator, resulting in a ramp-type drift with the slope of .3 units/sec.

It is believed that the first two components are responsible for the "main" dynamics of the azimuth channel, and can be defined as

$$G_{XX1}(s)=32/(s+200),$$

and

$$G_{XX2}(s)=1.444e7/(s^2+3800s+1.444e7),$$

The third term represents a long-lasting high frequency vibration which propagates through the mirror structure (bending mode) and does not results in any residual effects. It can be defined as

$$G_{XX3}(s)=183.75s/(s^2+210s+1.225e7),$$

The drift term, clearly seen on the recorded step response curves, can be defined as

$$G_{XX4}(s)=.3/s,$$

Finally, mathematical model of the azimuth control channel of the 500Hz mirror (closed-loop) is

$$G_{XX}(s)=G_{XX1}(s)+G_{XX2}(s)+G_{XX3}(s)+G_{XX4}(s).$$

The simulated step responses, given in Figures 3-1 and 3-3, show very good resemblance of the experimental results.

Figure 3-2 features the frequency response of the X-to-X channel which exhibits noticeable nonlinearity within the operational frequency range, and a resonant peak at the frequency of bending modes.

Mathematical description and simulation of the closed-loop transfer function of the elevation channel, $G_{YY}(s)$, is very consistent with $G_{XX}(s)$ and was also quite successful:

$$G_{YY}(s)=G_{YY1}(s)+G_{YY2}(s)+G_{YY3}(s)+G_{YY4}(s),$$

where

$$G_{YY1}(s)=32/(s+200),$$
$$G_{YY2}(s)=1.444e7/(s^2+2800s+1.444e7),$$
$$G_{YY3}(s)=183.75s/(s^2+210s+1.225e7),$$
$$G_{YY4}(s)=.3/s,$$

Mathematical description of the crosscoupling channels was obtained in the form,

$$G_{XY}(s)=G_{YX}(s)=150.s/(s^2+210s+1.225e7).$$

Figure 3-1. Step response of the math.model of the X-X channel

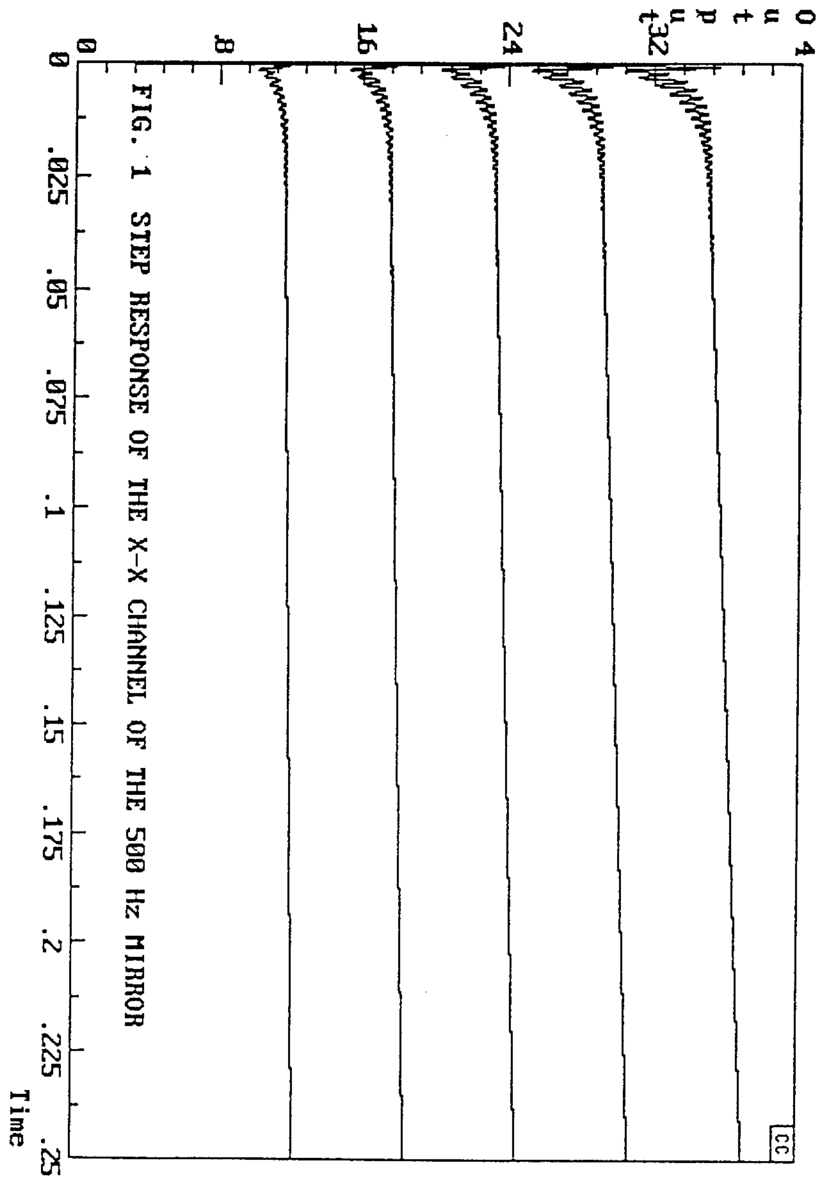


Figure 3-2. Estimated frequency response of the X-X channel

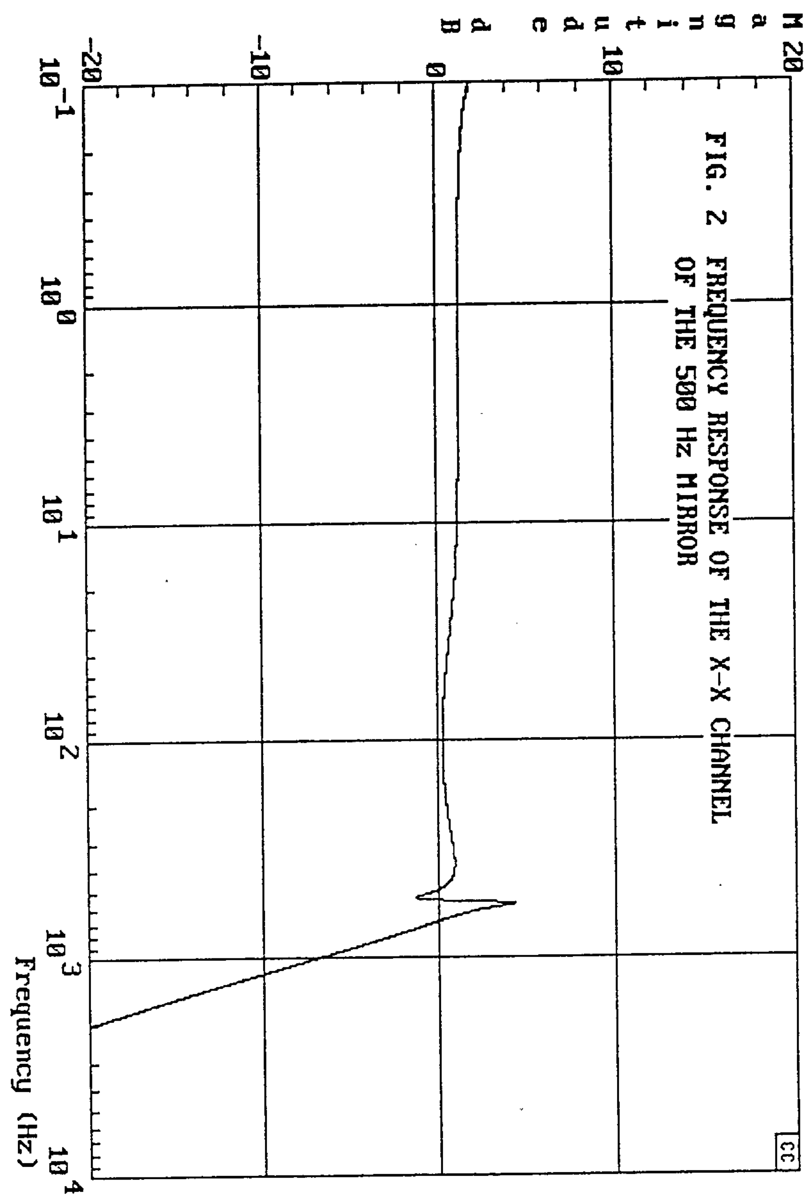
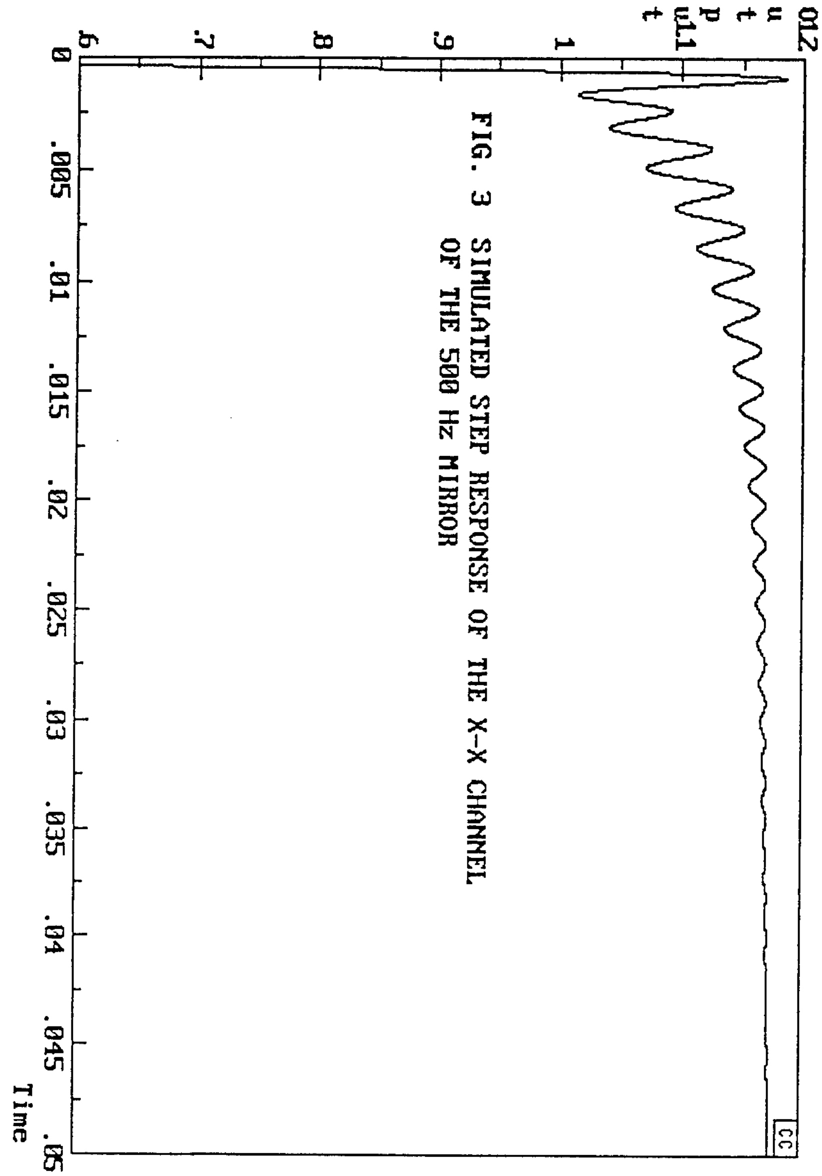


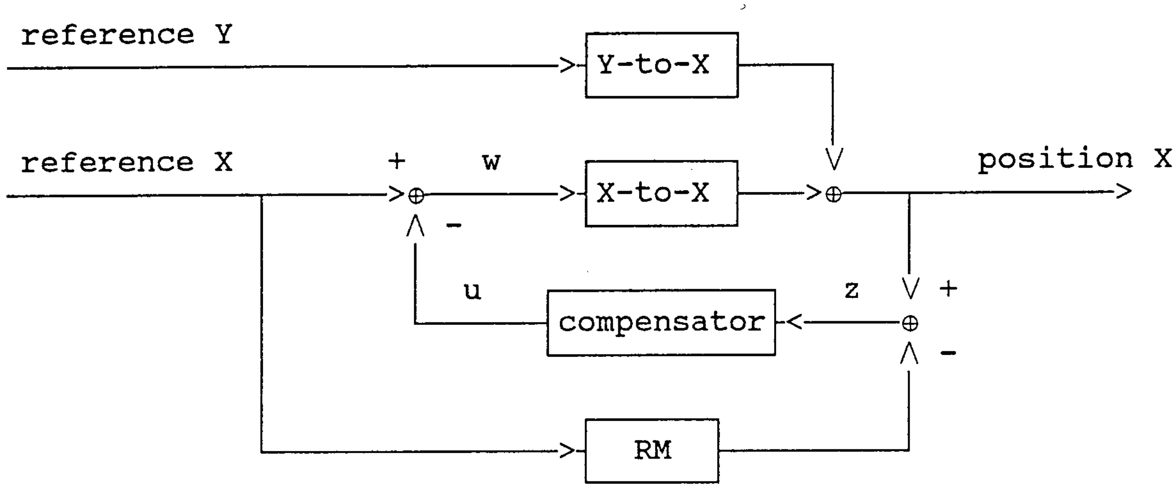
Figure 3-3. Step response of the math.model of the X-Y channel



4. APPLICATION OF THE MODEL REFERENCE APPROACH FOR PERFORMANCE MODIFICATION OF THE 500Hz MIRROR

Availability of mathematical models, describing the "main" dynamics of the X-to-X, Y-to-Y, X-to-Y, and Y-to-X mirror control channels, facilitates the implementation of the model reference (MR) control scheme, eliminating undesirable components of mirror dynamics, such as bending modes and crosscoupling.

The block diagram below illustrates the principle of operation of the MR scheme.

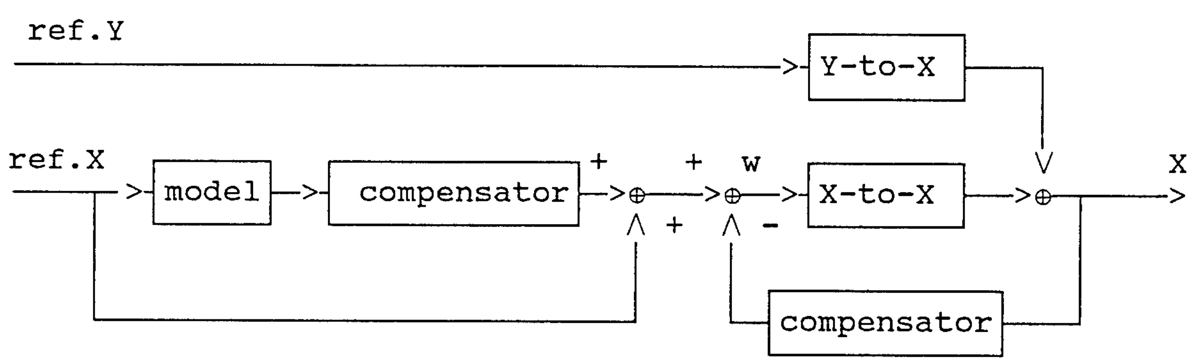


Blocks X-to-X and Y-to-X represent dynamics of the "direct" mirror channel, and crosscoupling between channels. Block "RM" (reference model) represents the "main" dynamics of the appropriate channel, free of bending modes and effects of crosscoupling. It could be seen that error signal $z(t)$ represents all undesirable motion components (including bending modes and effects of crosscoupling), which are extracted by the error detector. Block "compensator" is responsible for the transformation of the detected error signal into compensation signal $u(t)$, injected in the reference path of the mirror control circuitry.

Design of the MR system is aimed at the selection of such compensation law, $H(s)$, which would "force" the output of the mirror follow the output of the reference model. Since the

output of the reference model includes only the "main dynamics" terms, the compensator is expected to reduce all undesirable components of the system response such as bending modes and effects of crosscoupling. However, the benefits of a MR system do not come without a price: compensation signal u(t) results in the increased magnitude of the control efforts in the mirror positioning system. The increased control effort may or may not be carried out by the existing mirror drive devices, and may results in some additional explicit or implicit costs. Therefore, the feasibility of control effort is one of the important considerations during the design of a MR system. While the magnitude of control effort is of primary concern, computer simulations present the only opportunity for dealing with this issue.

It can be shown that the following is the equivalent representation of the above system, more suitable for computer simulation analysis and design of the mirror control circuitry:



The model reference control scheme, considered above, was developed for the 500 Hz optical mirror and implemented in the form of a simulation model. The following first order system was selected as the reference model:

$$G_{MODEL}(s)=4680/(s+4000)$$

It was found that the most suitable controller H(s) can be defined in the form of a phase-lead circuit,

$$H(s)=(1+.001s)/(1+.0001s)$$

Step response of the X-X channel, after adding the compensation scheme, is shown in Figure 4-1. It can be seen that the response is virtually free from the high frequency "bending mode" terms. Frequency response of the X-X channel is shown in Figure 4-2. It can be seen that application of the MR approach has resulted in the enhanced operational characteristics of the 500Hz mirror: its bandwidth is almost doubled, and the magnitude of the resonant peak, corresponding to bending modes, is drastically reduced.

Recall that in the original simulation tests the magnitude of the input signal, applied to the mirror circuitry, was equal to 1.0. Figure 4-3 presents the control effort, generated in the model reference compensation scheme: its magnitude is equal to 2., i.e. the "cost" of the performance enhancement is doubling the magnitude of the control effort.

Application of the reference model

$$G_{MODEL}(s)=7020/(s+6000)$$

results in the appropriate reduction of the settling time, elimination of high frequency transient terms, and extends the system bandwidth to 1.5KHz with the increase of the control effort by 200%. (See Figures 4-4 and 4-5).

In theory, a model reference compensation scheme is capable of the elimination/reduction of all "undesirable" components of the transient response, including the ones caused by cross-coupling.

A simulation study of the decoupling effect of the model reference scheme was conducted. The block diagram of the equivalent simulation setup is shown below:

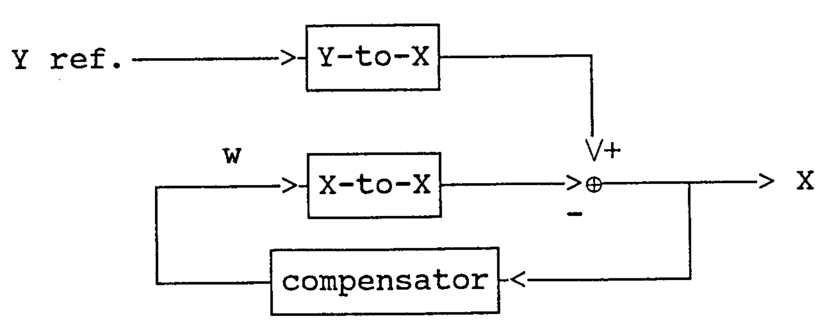


Figure 4-1.

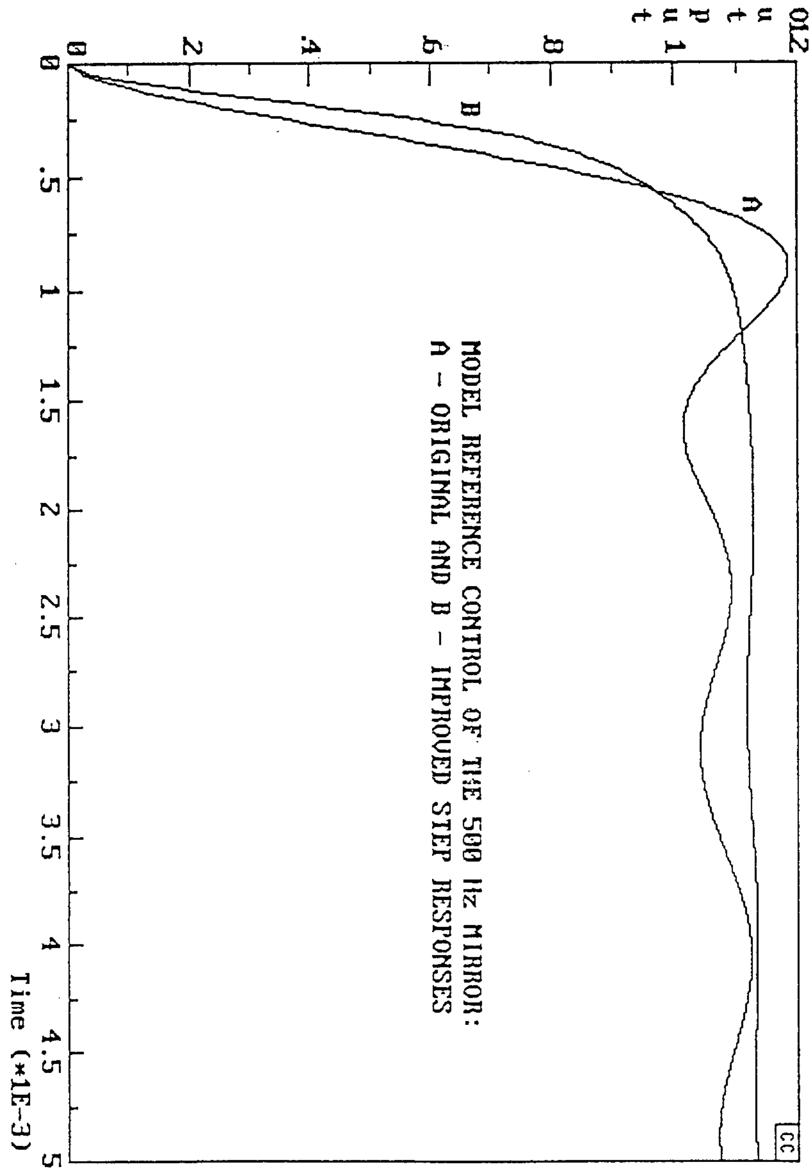
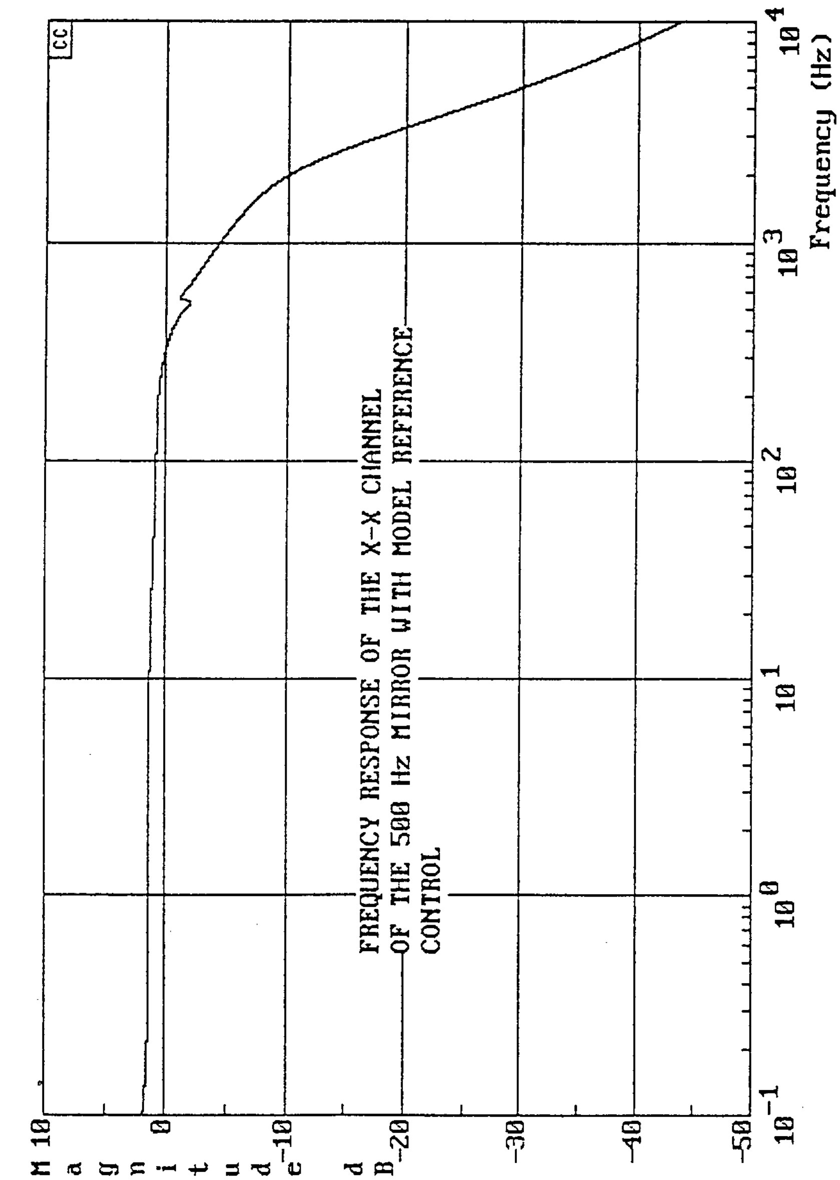
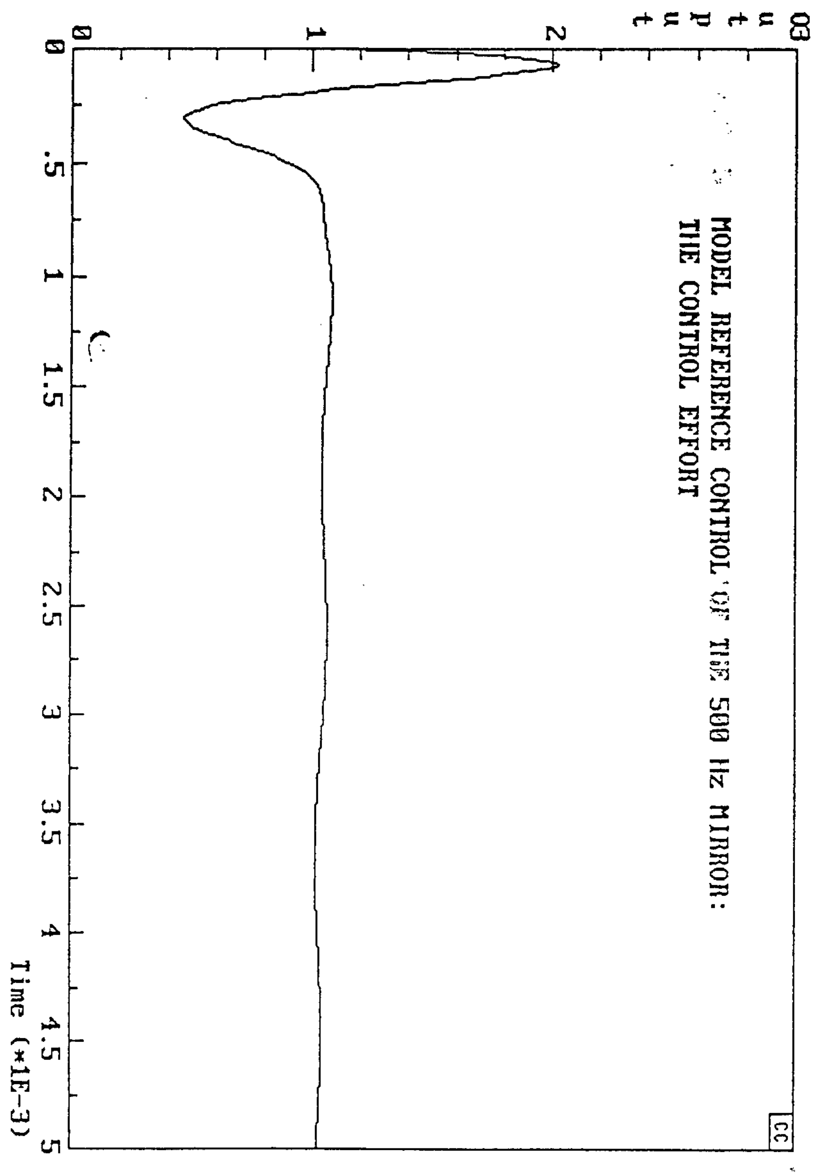

MODEL REFERENCE CONTROL OF THE 500 Hz MIRROR:
A - ORIGINAL AND B - IMPROVED STEP RESPONSES
Output
Time (*1E-3)
A
B
CC

Figure 4-2.



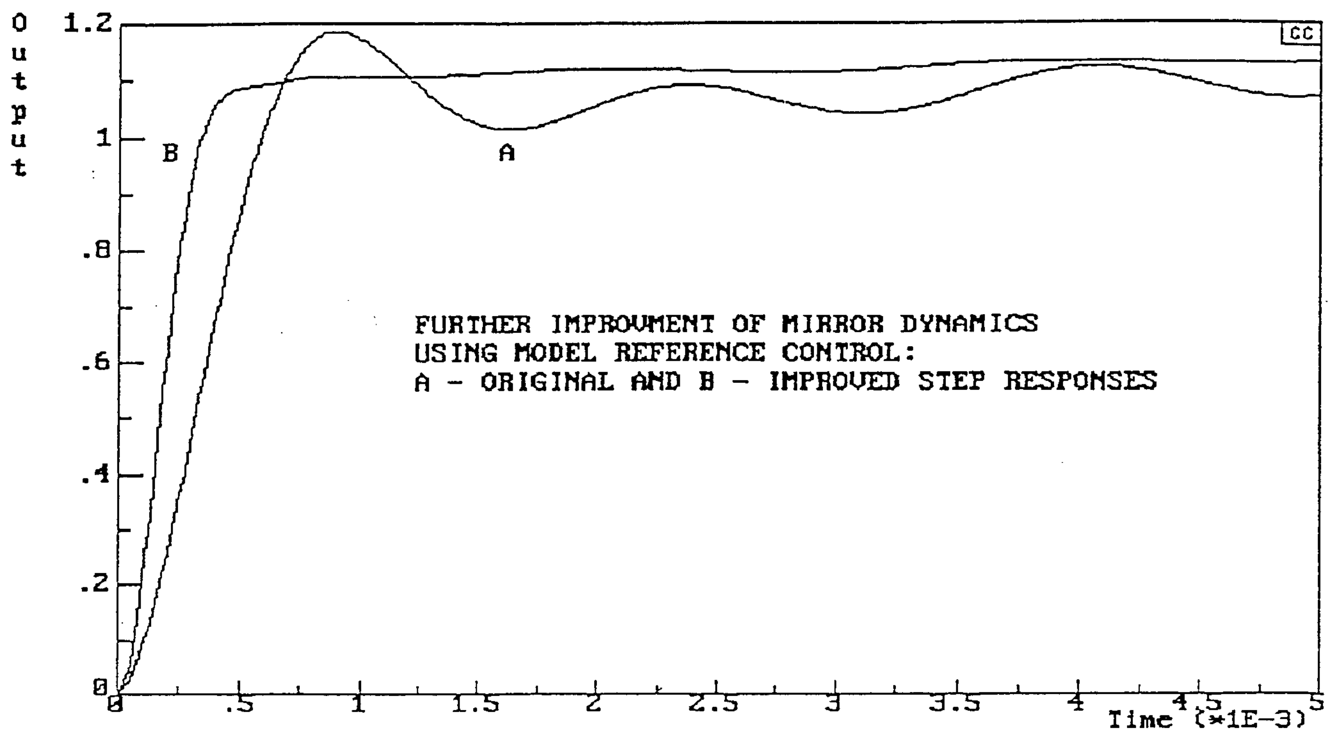
FREQUENCY RESPONSE OF THE X-X CHANNEL
OF THE 500 Hz MIRROR WITH MODEL REFERENCE
CONTROL
Magnitude dB
10
0
-10
-20
-30
-40
-50
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (Hz)
CC

Figure 4-3.



MODEL REFERENCE CONTROL OF THE 500 Hz MIRROR:
THE CONTROL EFFORT
Output
Time (*1E-3)
0
.5
1
1.5
2
2.5
3
3.5
4
4.5
5
0
1
2
3
CC

Figure 4-4.


Output
1.2
1
.8
.6
.4
.2
0
B
A
FURTHER IMPROVMENT OF MIRROR DYNAMICS
USING MODEL REFERENCE CONTROL:
A - ORIGINAL AND B - IMPROVED STEP RESPONSES
0
.5
1
1.5
2
2.5
3
3.5
4
4.5
5
Time (*1E-3)
CC

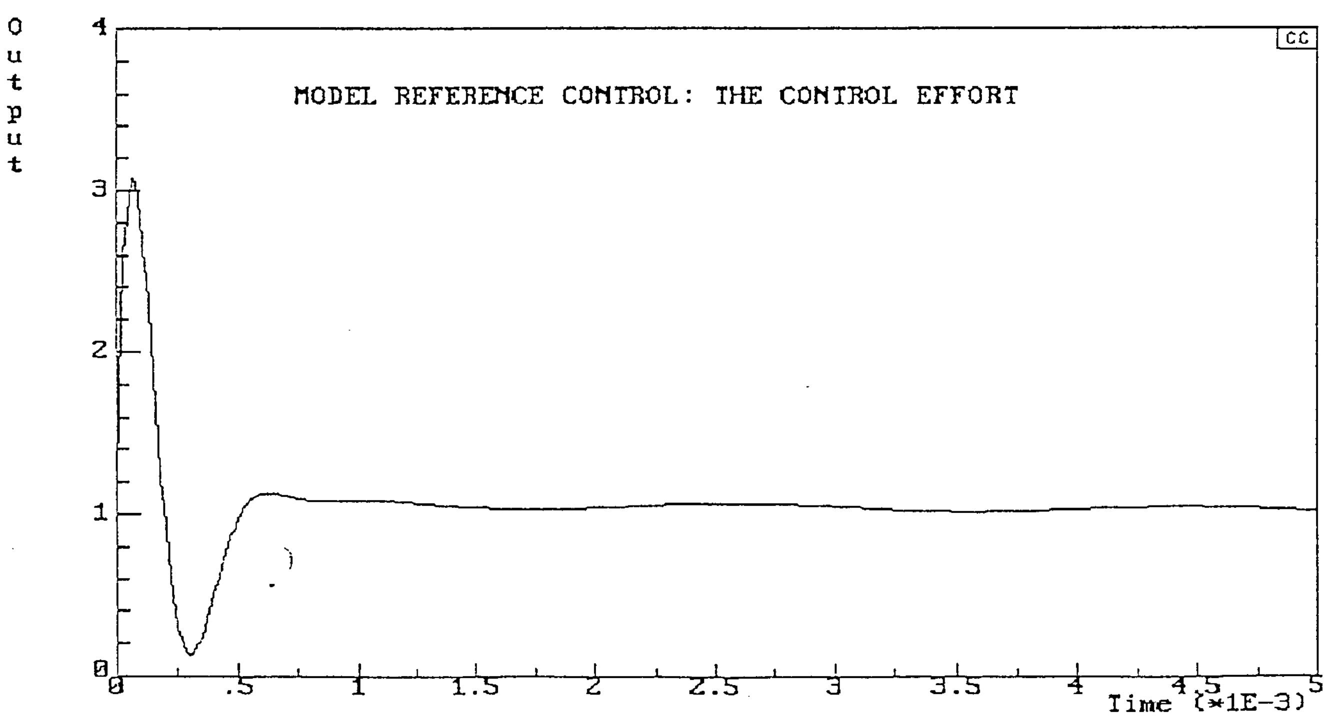

Output
4
3
2
1
0
MODEL REFERENCE CONTROL: THE CONTROL EFFORT
0
.5
1
1.5
2
2.5
3
3.5
4
4.5
5
Time (*1E-3)
CC

Figure 4-5.

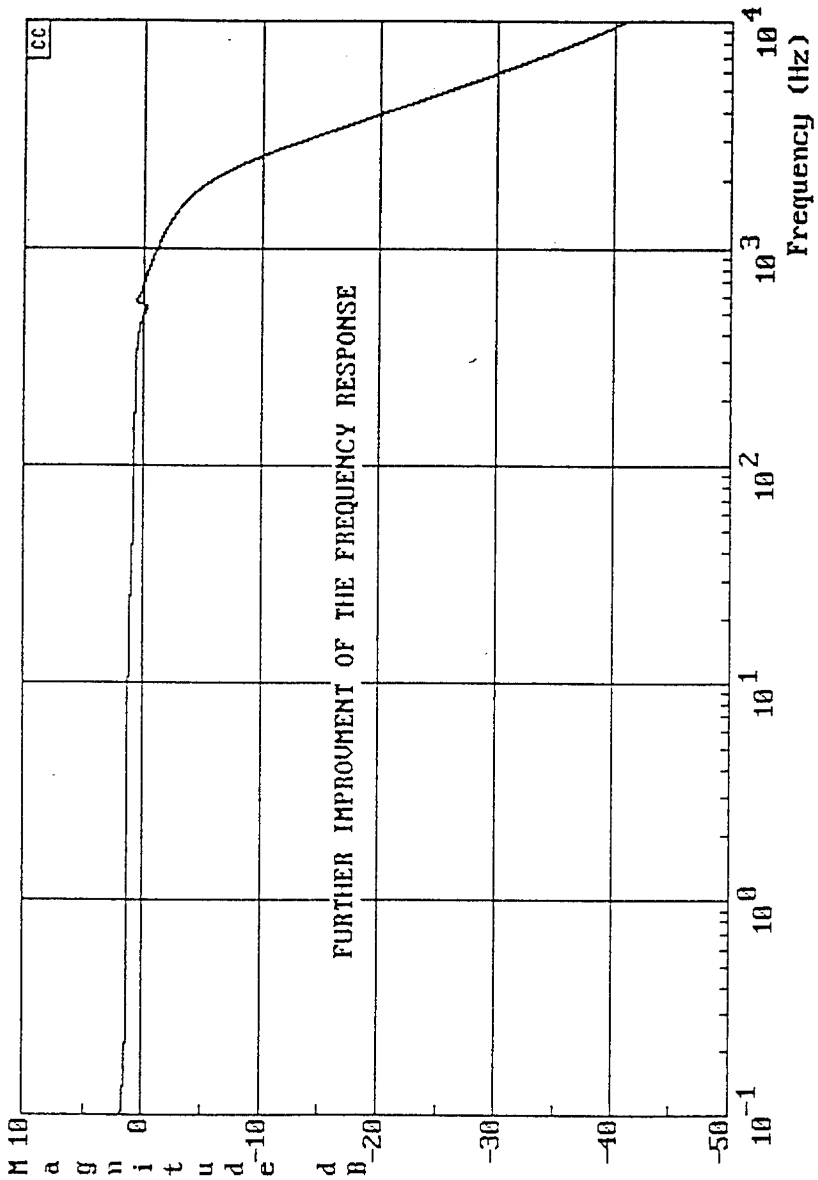
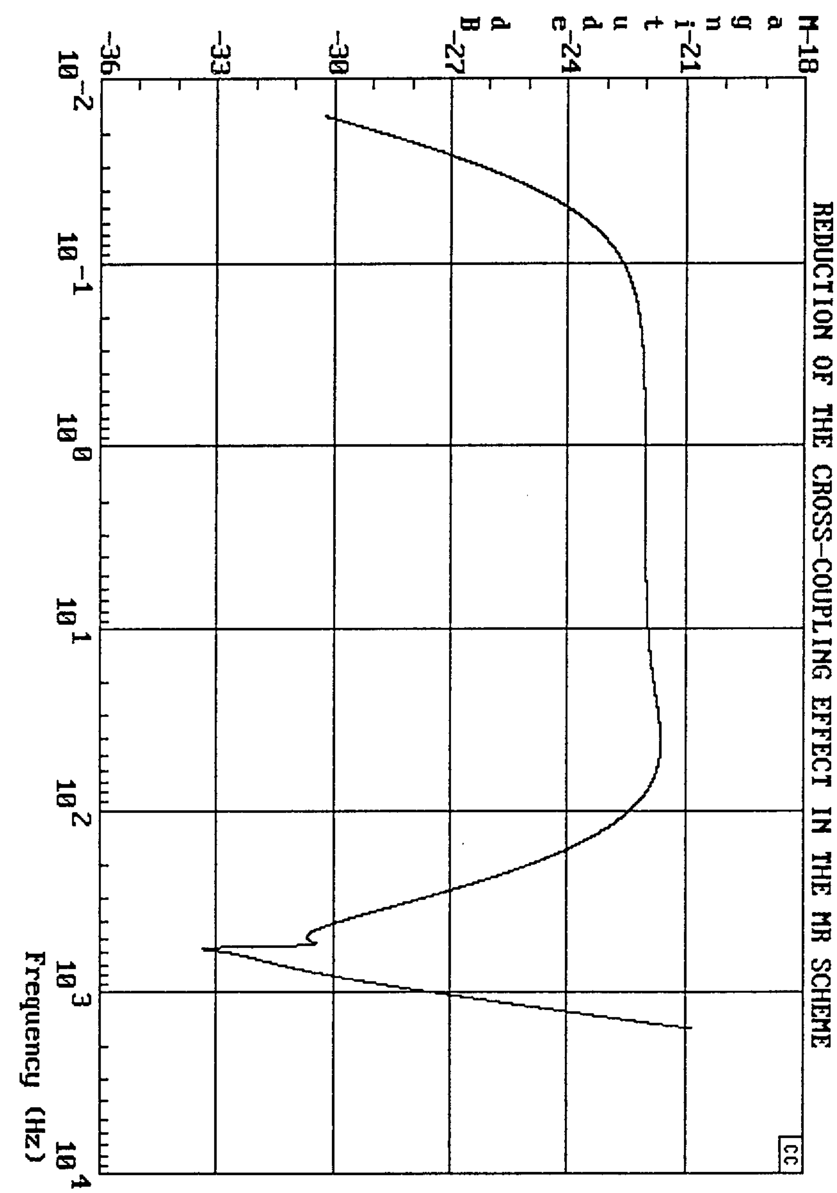

FURTHER IMPROVMENT OF THE FREQUENCY RESPONSE
Frequency (Hz)
Magnitude dB
CC
10
0
-10
-20
-30
-40
-50
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}

Figure 4-6.


REDUCTION OF THE CROSS-COUPLING EFFECT IN THE MR SCHEME
Magnitude dB
-18
-21
-24
-27
-30
-33
-36
10^-2
10^-1
10^0
10^1
10^2
10^3
10^4
Frequency (Hz)
CC

It could be seen that reduction of cross-coupling effects depends on the frequency response of the transfer function with unity forward path and the feedback

$$G_{FB}(s)=G_{XX}(s)H(s)=$$
$$[32/(s+200)+1.444e7/(s^2+3800s+1.444e7)+$$
$$+83.75s/(s^2+210s+1.225e7)+.3/s][(1+.001s)/(1+.0001s)]$$

The appropriate frequency response indicates that only modest reduction of cross-coupling effect (5 db) can be expected. However, this reduction can be achieved by increasing the gain of the compensator H(s). For example, Figure 5-6 indicates that at

$$H(s)=10(1+.001s)/(1+.0001s)$$

cross-coupling effect can be reduced by 20 db within the frequency range of interest. According to simulation analysis, the compensator gain can be increased up to 100 without any stability loss or noticeable increase of the magnitude of control efforts.

The implementation of the MR control technology requires that the model reference and the compensator be properly defined in software or hardware. The following discrete-time expressions can be utilized for the software implementation.

Reference model via the central difference scheme:

$Y(s)/X(s)=k/(s+a)$, or

$sY(s)+aY(s)=kX(s)$, or

$y'(t)+ay(t)=kx(t)$, or

$y[(i+1)\Delta^t]-y[(i-1)\Delta^t]+2a\Delta^t y[i\Delta^t]=2k\Delta^t x[i\Delta^t]$, or

$y_{i+1}-y_{i-1}+2a\Delta^t y_i=2k\Delta^t x_i$, or

$y_{i+1}=y_{i-1}-2a\Delta^t y_i+2k\Delta^t x_i$,

where

$y(t)$, $x(t)$, $Y(s)$, and $X(s)$ are output and input signals of the reference model and their Laplace transforms,

k and a are parameters of the reference model,

t - continuous time,

Δ^t is time discretization step, and

$y[i\Delta^t]=y_i=y(t)/t=i\Delta^t$ and $x[i\Delta^t]=x_i=x(t)/t=i\Delta^t$,

The compensator via the central difference scheme (using the same notations):

$Y(s)/X(s)=k(1+as)/(1+bs)$, or

$sbY(s)+Y(s)=ksaX(s)+kX(s)$, or

$by'(t)+y(t)=kax'(t)+kx(t)$, or

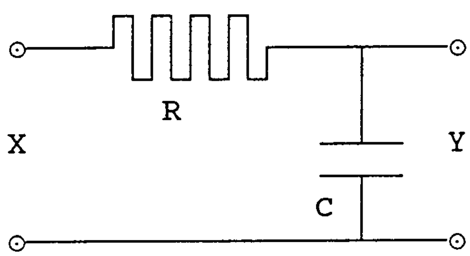
$by_{i+1}-by_{i-1}+2\Delta^t y_i=kax_{i+1}-kax_{i-1}+2k\Delta^t x_i$, or

$by_i-by_{i-2}+2\Delta^t y_{i-1}=kax_i-kax_{i-2}+2k\Delta^t x_{i-1}$, or

$y_i=[by_{i-2}-2\Delta^t y_{i-1}+kax_i-kax_{i-2}+2k\Delta^t x_{i-1}]/b$

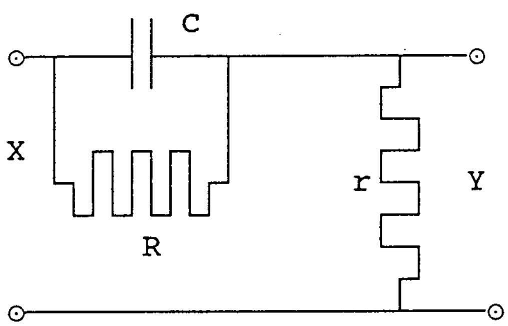
Implementation of the MR control using analog circuitry:

- first order system



$$\frac{Y(s)}{X(s)}=\frac{1}{RCs+1}$$

- phase-lead system



$$\frac{Y(s)}{X(s)}=\frac{s+1/RC}{s+(R+r)/RrC}$$

CONCLUSION: Characteristics of high performance optical mirrors can be significantly improved by upgrading their existing control systems using model reference control schemes:

- frequency response can be extended and linearized within the operational frequency range,

- high frequency components of the transient response (bending modes) can be eliminated,

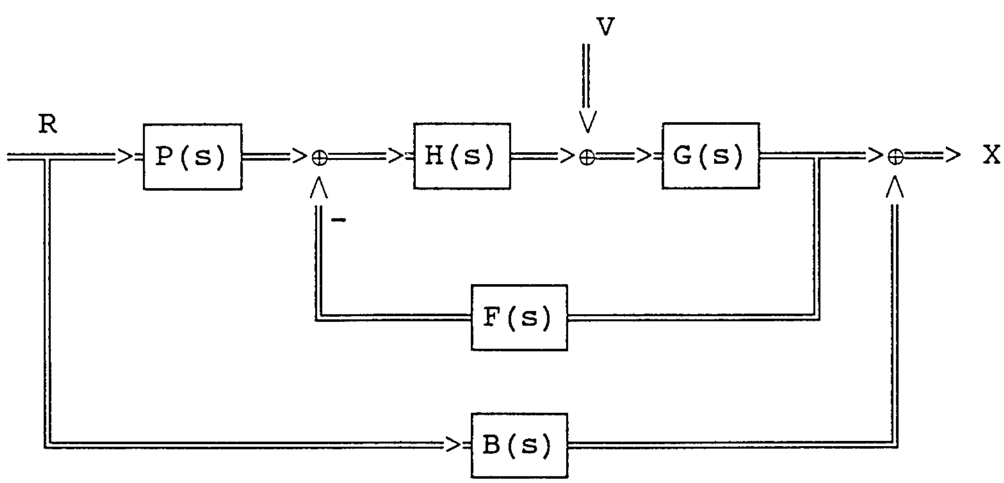
- the cross-coupling effect can be controlled.

The potential of the MR approach is limited only by the allowable magnitude of control effort.

5. MATHEMATICAL DESCRIPTION AND COMPUTER SIMULATION OF A HIGH PRECISION POSITION CONTROL SYSTEM

This section of the Report presents a simulation study on various aspects of the application of the Model Reference (MR) approach in high precision positioning systems. While in the previously conducted studies, the original positioning system was defined by a single dynamic block representing "closed-loop" system behavior, it is assumed herein that the entire block diagram of the original system and its particular modules are known. This assumption allows for the explicit and accurate investigation of control efforts in the original and MR systems, system sensitivity to jitter, effects of cross-coupling, and system sensitivity to internal parameters of the controlled plant. Note, that although the configuration of the system described below is typical of a fine steering mirror control system, the study is concerned with general principles and does not target any specific beam steering technology.

Consider a high precision position control system represented by the block diagram below.



The control loop of this system is comprised of electronic and electro-mechanical elements. The system dynamics includes high-frequency components of the transient process, so-called

bending modes. Presently, bending mode components are not "covered" by the existing position sensor. They are not present in the feedback signal, and are not compensated for by the existing controller.

Note that double lines symbolize vector signals, and blocks represent transfer matrices,

vector $\mathbf{X}=[X_1,X_2]^T$ represents the azimuth and elevation components of the position (^{T} is the transpose symbol),

vector $\mathbf{R}=[R_1,R_2]^T$ represents the azimuth and elevation components of the position reference,

signal V is a scalor representing the vibration (jitter) affecting the system,

transfer matrix $\mathbf{G}(s)$ represents the dynamics of the controlled process comprised of "direct channel" transfer functions $g_{11}(s)$ and $g_{22}(s)$, transfer functions $g_{12}(s)=g_{21}(s)$ representing crosscoupling effects, and transfer functions $g_{13}(s)=g_{23}(s)$ representing effects of vibration.

$$\mathbf{G}(s)=\begin{bmatrix} g_{11}(s) & g_{12}(s) & g_{13}(s) \\ g_{21}(s) & g_{22}(s) & g_{23}(s) \end{bmatrix}$$

Transfer matrix $\mathbf{H}(s)$ represents the combined dynamics of the system's drive generating the azimuth and elevation control efforts, U_1, and U_2, and the prefilter placed in the system's forward path:

$$\mathbf{H}(s)=\begin{bmatrix} h_{11}(s) & 0 \\ 0 & h_{22}(s) \end{bmatrix}$$

Transfer matrix $\mathbf{B}(s)$ represents the effect of "bending modes" on the position output,

$$\mathbf{B}(s)=\begin{bmatrix} b_{11}(s) & 0 \\ 0 & b_{22}(s) \end{bmatrix}$$

Transfer matrix **F**(s) represents the feedback controllers,

$$F(s)=\begin{bmatrix} f_{11}(s) & 0 \\ 0 & f_{22}(s) \end{bmatrix}$$

Transfer matrix **P**(s) represents the prefilters in the reference channel,

$$\mathbf{P}(s)=\begin{bmatrix} p_{11}(s) & 0 \\ 0 & p_{22}(s) \end{bmatrix}$$

For the purpose of simulation, the following definitions of the particular transfer functions were selected:

$g_{11}(s)=g_{22}(s)=(s+2)/[s(s+20)]$

$b_{11}(s)=100s/(s^2+36s+600^2)$

$b_{22}(s)=126s/(s^2+112s+800^2)$

$g_{12}(s)=2.3s/(s^2+30s+150^2)$

$g_{13}(s)=310/(s^2+12.36s+3600)+66*s/(s^2+9.6s+600^2)$

$h_{11}(s)=h_{22}(s)=(s+20)/(s+40)$

$f_{11}(s)=f_{22}(s)=1000/(s+20)$

$p_{11}(s)=p_{22}(s)=1000/(s+20)$

On the basis of these definitions and system configuration, the following transfer functions of interest can be established:

- transfer function of the existing system (reference-to-position):

$G_E(s)=p_{11}(s)G_0(s)+b_{11}(s)$

where

$$G_0(s)=\frac{h_{11}(s)g_{11}(s)}{1+h_{11}(s)g_{11}(s)f_{11}(s)}$$

- transfer function of the existing system (reference-to-control effort):

$$G_{EU}(s)=\frac{p_{11}(s)h_{11}(s)}{1+h_{11}(s)g_{11}(s)f_{11}(s)}$$

- transfer function of the existing system (jitter-to-position):

$$G_{EJIT}(s)=\frac{g_{13}(s)}{1+h_{11}(s)g_{11}(s)f_{11}(s)}$$

- transfer function of the existing system describing the cross-coupling effects (reference-to-position of the adjacent channel):

$$G_{ECC}(s)=\frac{p_{22}(s)h_{22}(s)}{1+h_{22}(s)g_{22}(s)f_{22}(s)}*\frac{g_{12}(s)}{1+h_{11}(s)g_{11}(s)f_{11}(s)}$$

The following graphs represent simulation results describing the performance of the described positioning system. It should be noted that the existing controller does not utilize the high frequency components of the position response, so-called "bending modes", which presents the major system drawback.

Figure 5-1 represents step and frequency responses of the "reference - position" channel.

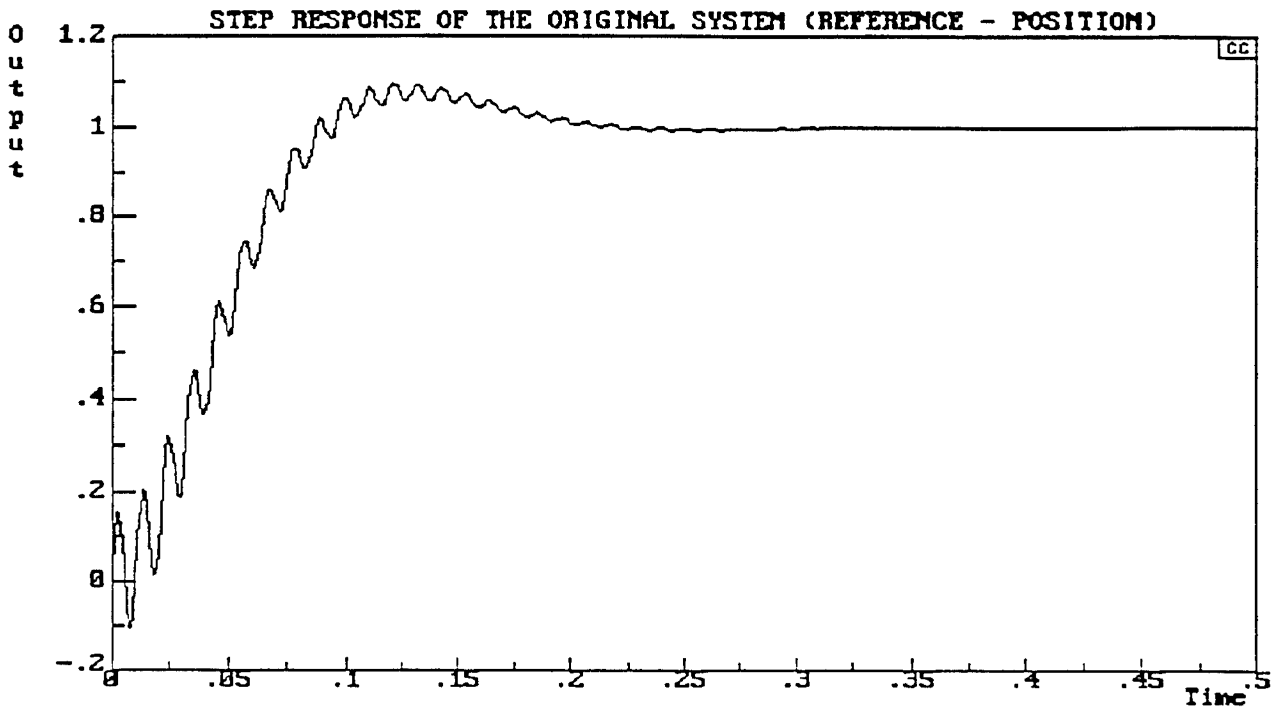
Figure 5-2 shows step and frequency responses of the "jitter - position" channel, representing system sensitivity to environmental vibrations.

Figure 5-3 features step and frequency responses of the "reference - position of the coupled channel" thus representing cross-coupling effect.

Figure 5-4 shows the control effort in the positioning system while the system responds to the unit step reference signal.

Figure 5-5 presents the family of step responses of the "reference - position" channel as the pole of the transfer function of the controlled plant, normally equal to -20, varies from -20 to -10. According to the Figure, system performance deteriorates due to wear of its mechanical components thus exhibiting system's sensitivity to parameters of the controlled plant.

Figure 5-1


STEP RESPONSE OF THE ORIGINAL SYSTEM (REFERENCE - POSITION)
Output
1.2
1
.8
.6
.4
.2
0
-.2
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time
CC

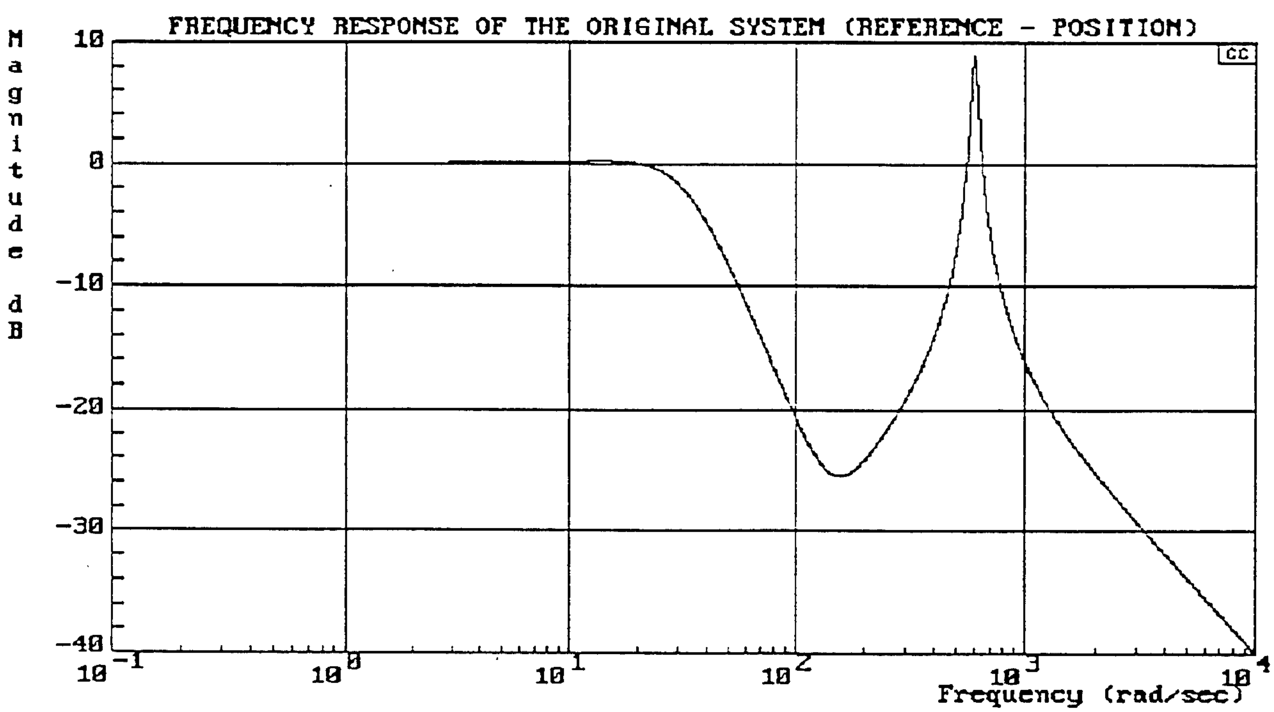
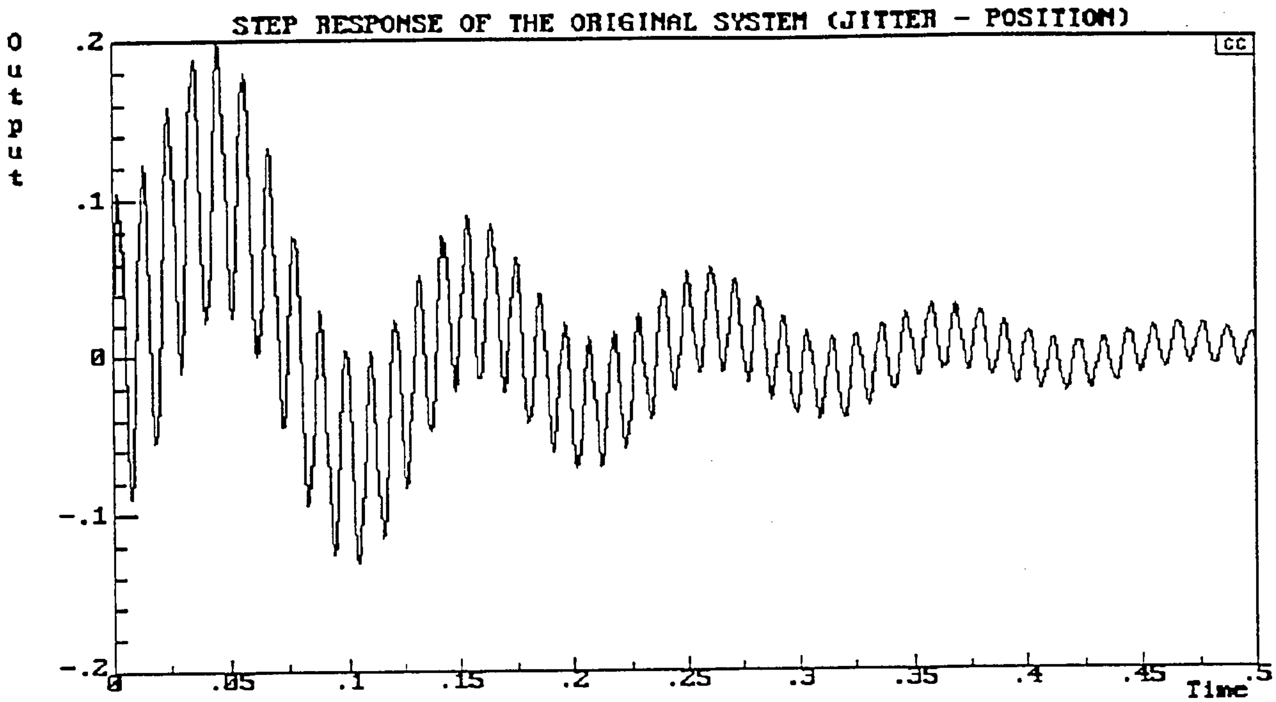

FREQUENCY RESPONSE OF THE ORIGINAL SYSTEM (REFERENCE - POSITION)
Magnitude dB
10
0
-10
-20
-30
-40
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)
CC

Figure 5-2


STEP RESPONSE OF THE ORIGINAL SYSTEM (JITTER - POSITION)
Output
.2
.1
0
-.1
-.2
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time

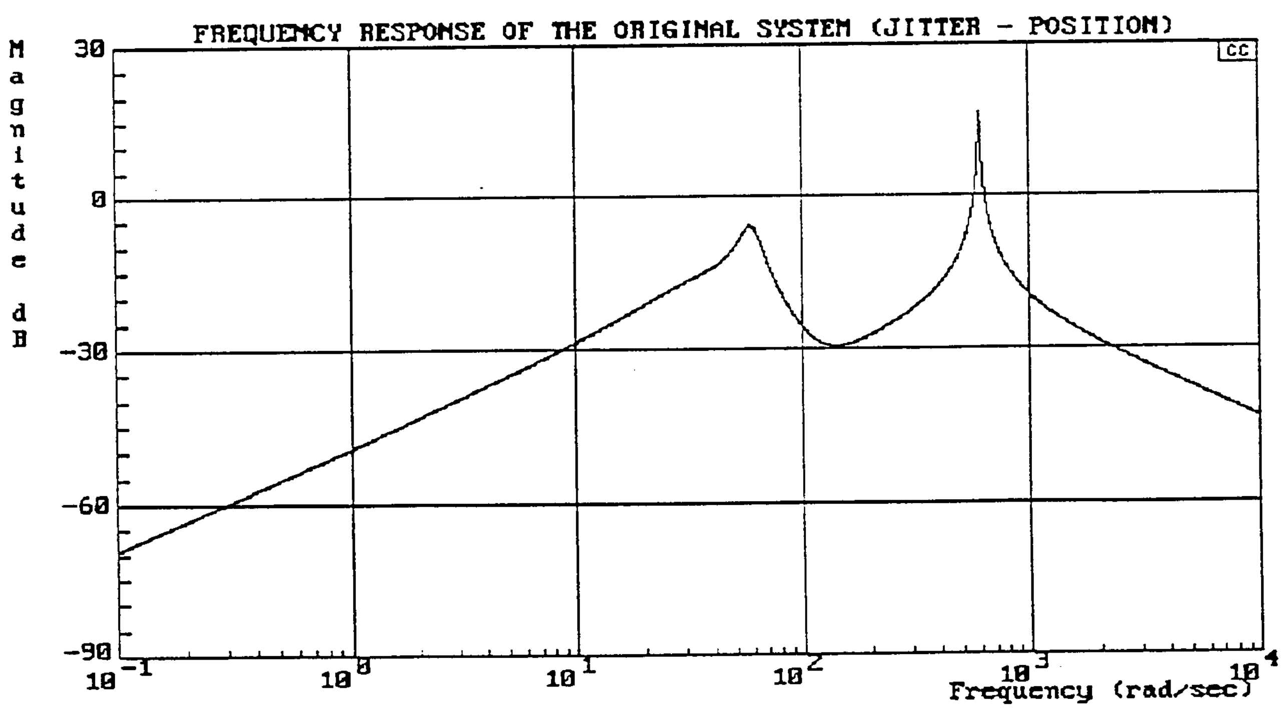
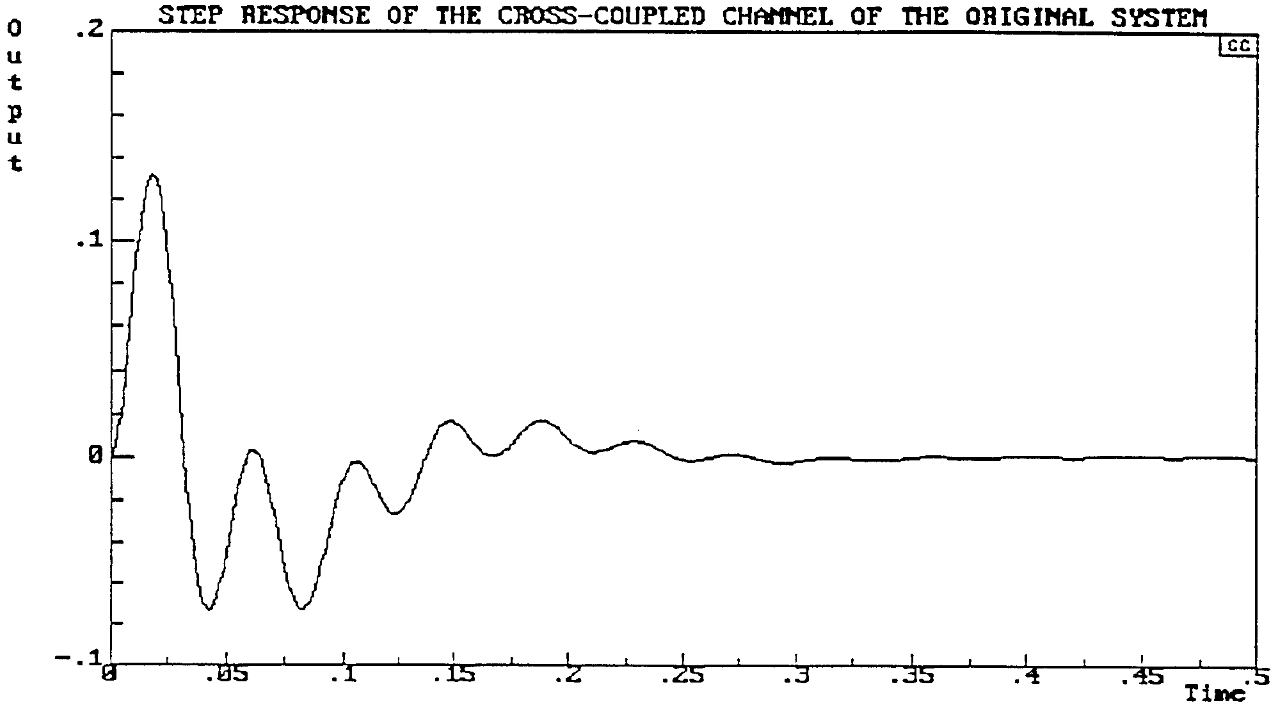

FREQUENCY RESPONSE OF THE ORIGINAL SYSTEM (JITTER - POSITION)
Magnitude dB
30
0
-30
-60
-90
10^-1
10^0
10^1
10^2
10^3
10^4
Frequency (rad/sec)

Figure 5-3


STEP RESPONSE OF THE CROSS-COUPLED CHANNEL OF THE ORIGINAL SYSTEM
CC
Output
.2
.1
0
-.1
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time

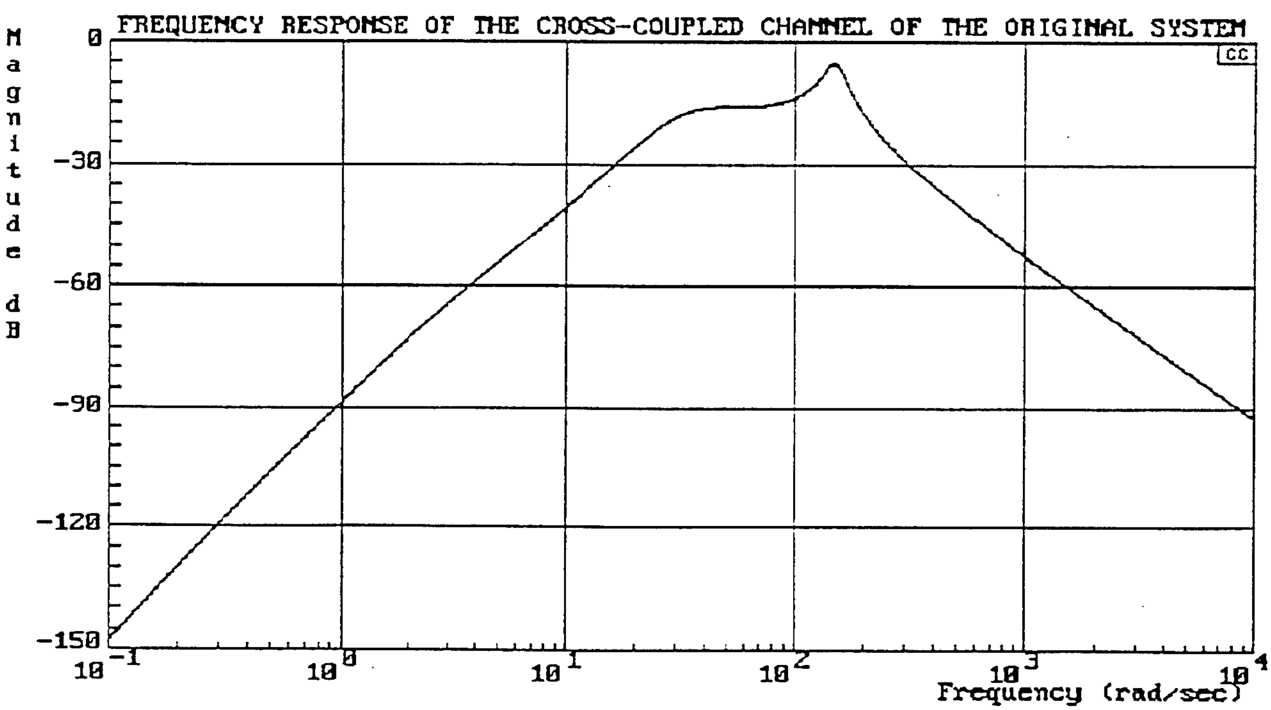
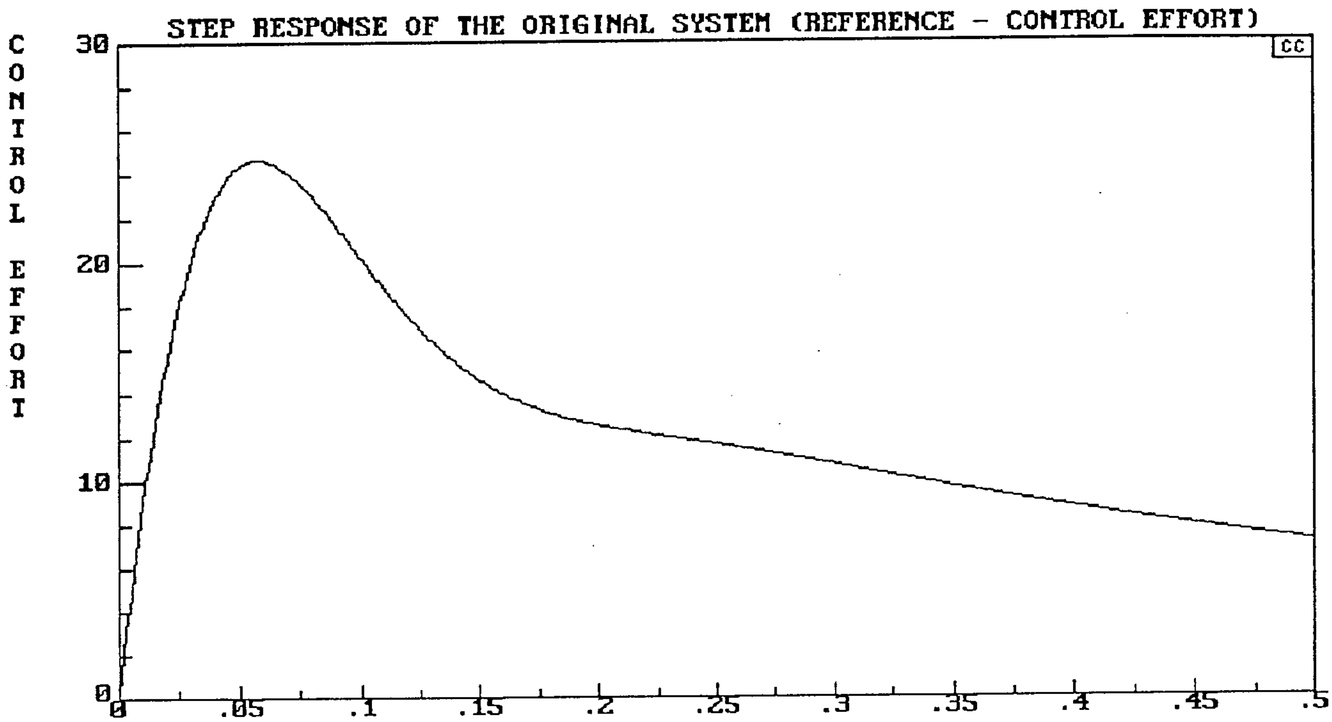
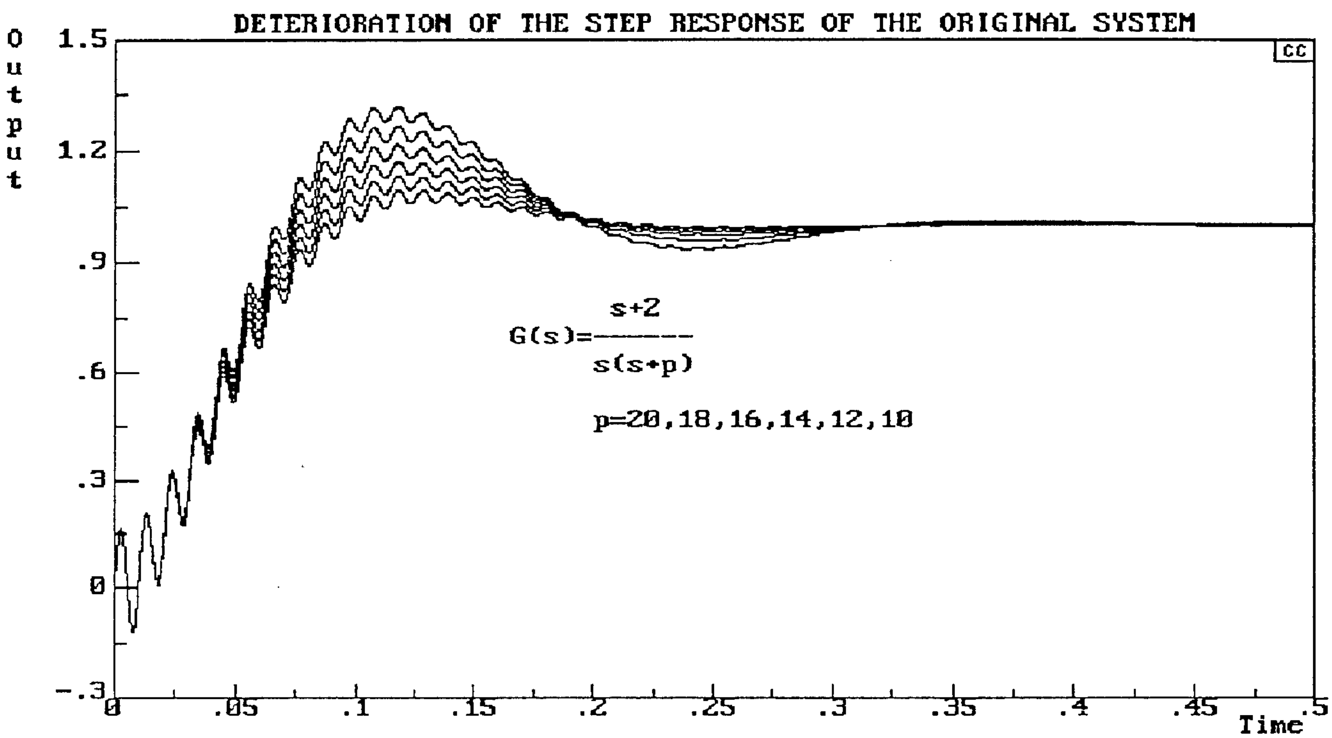

FREQUENCY RESPONSE OF THE CROSS-COUPLED CHANNEL OF THE ORIGINAL SYSTEM
CC
Magnitude dB
0
-30
-60
-90
-120
-150
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)

Figure 5-4



STEP RESPONSE OF THE ORIGINAL SYSTEM (REFERENCE - CONTROL EFFORT)
CC
CONTROL EFFORT
30
20
10
0
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5

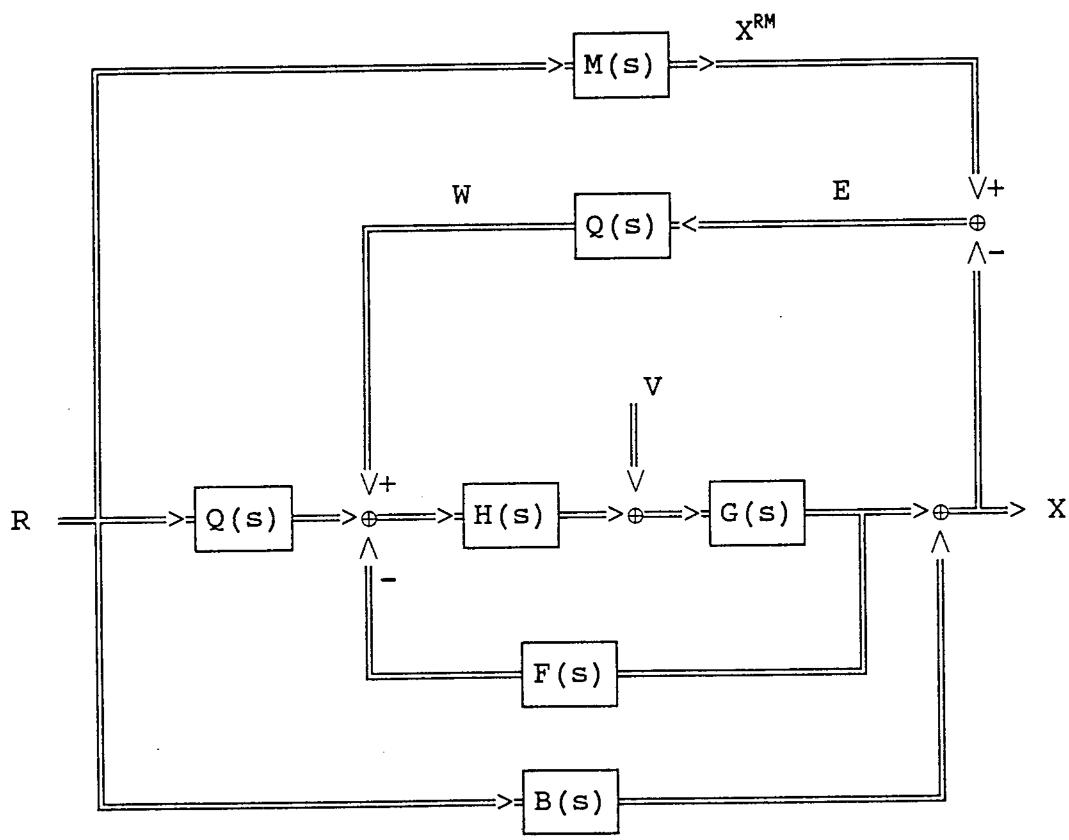
Figure 5-5



DETERIORATION OF THE STEP RESPONSE OF THE ORIGINAL SYSTEM
Output
1.5
1.2
.9
.6
.3
0
-.3
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time
CC
G(s)=s+2 / s(s+p)
p=20,18,16,14,12,10

6. APPLICATION OF THE MODEL REFERENCE APPROACH FOR THE PERFORMANCE ENHANCEMENT OF A HIGH PRECISION POSITION CONTROL SYSTEM

Consider the following configuration of the previously described control system, modified by adding a model reference (MR) loop. The reference model symbolized by block M(s), represents the desired behavior of the entire system. Block Q(s) is responsible for the generation of an additional control (adaptation) signal W, injected in the original control scheme.



Physically, the reference model is a computer simulation code, which cannot be affected by any undesirable conditions, such as cross-coupling between channels, parameter drift, and environmental jitter. The output of the reference model X^{RM}, therefore, represents the desired system response to the applied reference signal, reflecting only the dynamics of the reference model and the reference signal R. Unlike signal X^{RM}, output of

the control system X is dependent upon the reference signal R, dynamics of the "reference - position" channel of the system which, generally speaking, is different from the reference model, and the entire set of "real-life" conditions, such as cross-coupling between channels, parameter drift, and environmental jitter. Therefore, the MR approach utilizes the feedback control principle to extract the error signal $E=X^{RM}-X$ and to convert this error into a special adaptation signal $W=W(E)$. When properly defined, signal W, injected in the original control scheme, "forces" the output position X to follow the output of the reference model. As with any system utilizing the feedback control principle, the MR scheme provides a general purpose control action which assures that error $E=X^{RM}-X$ converges to zero, or stays within predictable limits, regardless of specific reasons causing the discrepancy between signals X^{RM} and X (difference in dynamics, parameter drift, cross-coupling, jitter).

Introduce transfer matrix **M**(s) representing the reference model,

$$\mathbf{M}(s)=\begin{bmatrix} m_{11}(s) & 0 & 0 \\ 0 & m_{22}(s) & 0 \end{bmatrix}$$

One can see that the reference model does not exhibit cross-coupling and cannot be affected by environmental jitter. The particular choice of transfer functions $m_{11}(s)=m_{22}(s)$ is given below:

$m_{11}(s)=100^2/(s^2+180s+100^2)$

Introduce transfer matrix **Q**(s) representing the adaptation rule,

$$\mathbf{Q}(s)=\begin{bmatrix} q_{11}(s) & 0 \\ 0 & q_{22}(s) \end{bmatrix}$$

While a general theory has been developed for the synthesis of MR control systems, specific adaptation rules can be successfully

selected following the guidelines of [12], and finalized by computer simulation. It was found that the following definitions of transfer functions Q(s) result in a stable control system, and a good match between the outputs of the reference model and the output of the control system:

$$q_{11}(s)=q_{22}(s)=1000(1+.1s)/(1+.01s)$$

The following graphs represent simulation results describing the performance of the synthesized positioning system with the conventional and MR control loops. It should be noted that while the conventional controller deals exclusively with the "main dynamics" of the controlled plant, the MR controller improves the "main dynamics" of the system, eliminates "bending modes", reduces cross-coupling, reduces effects of jitter, and increases the entire system robustness.

Figure 5-6 represents step and frequency responses of the selected reference model. It should be noted that dynamic characteristics of the reference model are expected to be "the same or better" than the ones of the original system. However, feasible magnitude of the control effort in the system presents the natural constraint on the choice of the reference model.

Figures 5-7 shows step and frequency responses of the "reference - position" channel of the MR system. Compared with the same characteristics of the original system (Figure 5-8), they indicate the possibility of significant enhancement of the system performance, providing that control effort (Figure 5-9) is attainable in the existing system.

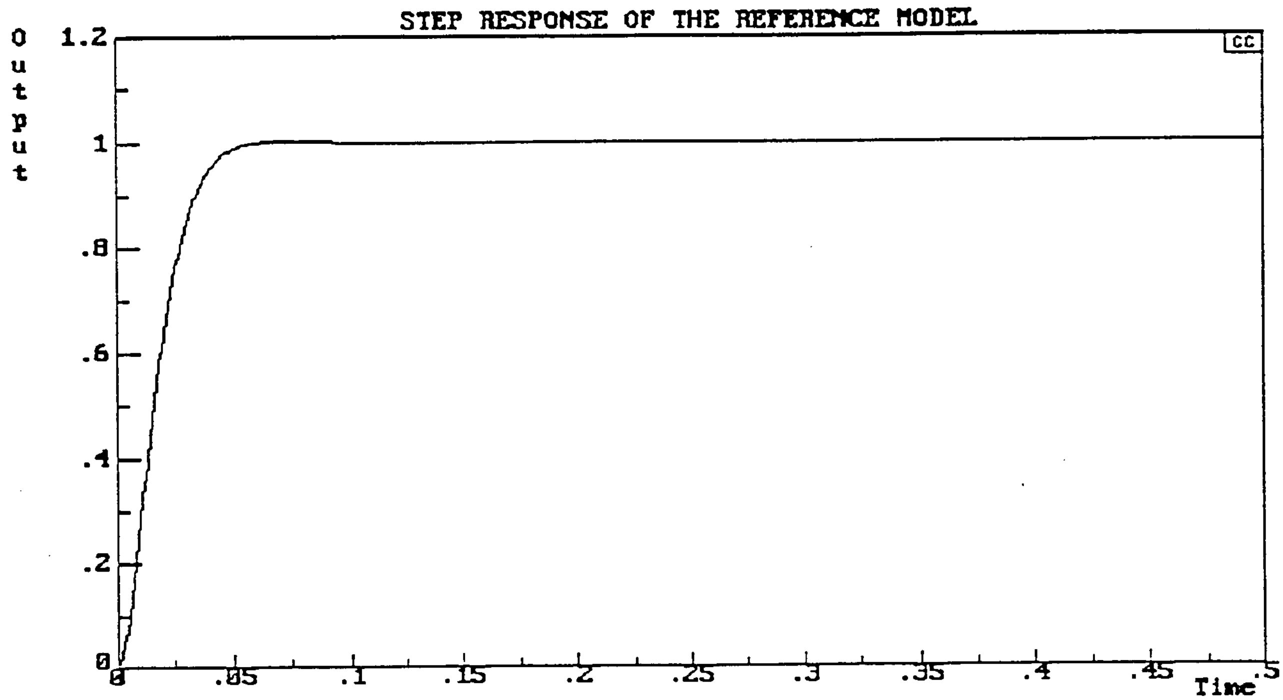
Figure 5-10 shows frequency response of the "jitter - position" channel of the MR system. Compared with the frequency response of the original system, it indicates a significant reduction (by 20 db) of system sensitivity to environmental vibrations within a particular range of frequencies ($10 - 10^4$ rad/sec). It is interesting that unlike adaptive feedforward scheme [2,3] which provides consistent jitter compensation effect in spite of frequency increase, the MR scheme follows the feedback principle and, therefore, is comparable with the

conventional controller at high frequencies. At low frequencies, due to particular choice of the adaptation law, it is not as efficient as the original feedback controller.

Figure 5-11 features step and frequency responses of the "reference - position of the coupled channel", representing cross-coupling effect in the MR system. Compared with the same characteristics of the original system (Figure 5-12) these results indicate that the MR scheme, generally speaking, improves system performance. However, at high frequencies frequency response of the MR system is consistently "worse" than the one of the original system. The explanation of this phenomenon is quite interesting. Note that transfer functions $g_{12}(s)$ and $g_{21}(s)$ relate position outputs of particular channels to the control efforts of the coupled channels. In order to compensate for "bending modes", the control effort of the MR system contains high frequency terms, see Figure 5-9. Therefore, even if channel 1 is affected by a step-type input signal, transient response in channel 2 is excited by the high-frequency control effort of channel 1. This results in the high-frequency resonant peak in the frequency response of the MR system.

Figure 5-13 presents the family of step responses of the "reference - position" channel in the MR system as the pole of the transfer function of the controlled plant varies from -20 to -10. According to the Figure, application of the MR scheme drastically improves system robustness, making it virtually insensitive to parameter drift of the controlled plant.

Figure 5-6


STEP RESPONSE OF THE REFERENCE MODEL
Output
1.2
1
.8
.6
.4
.2
0
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time

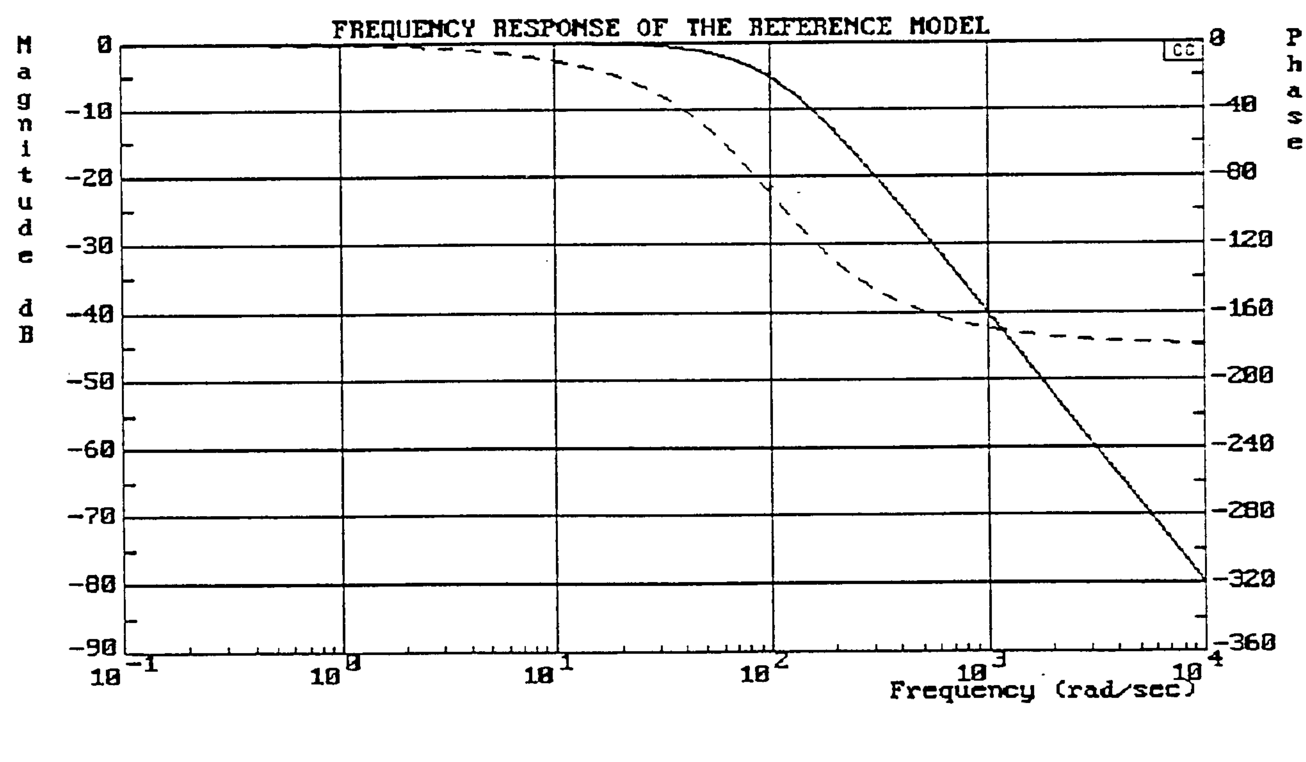
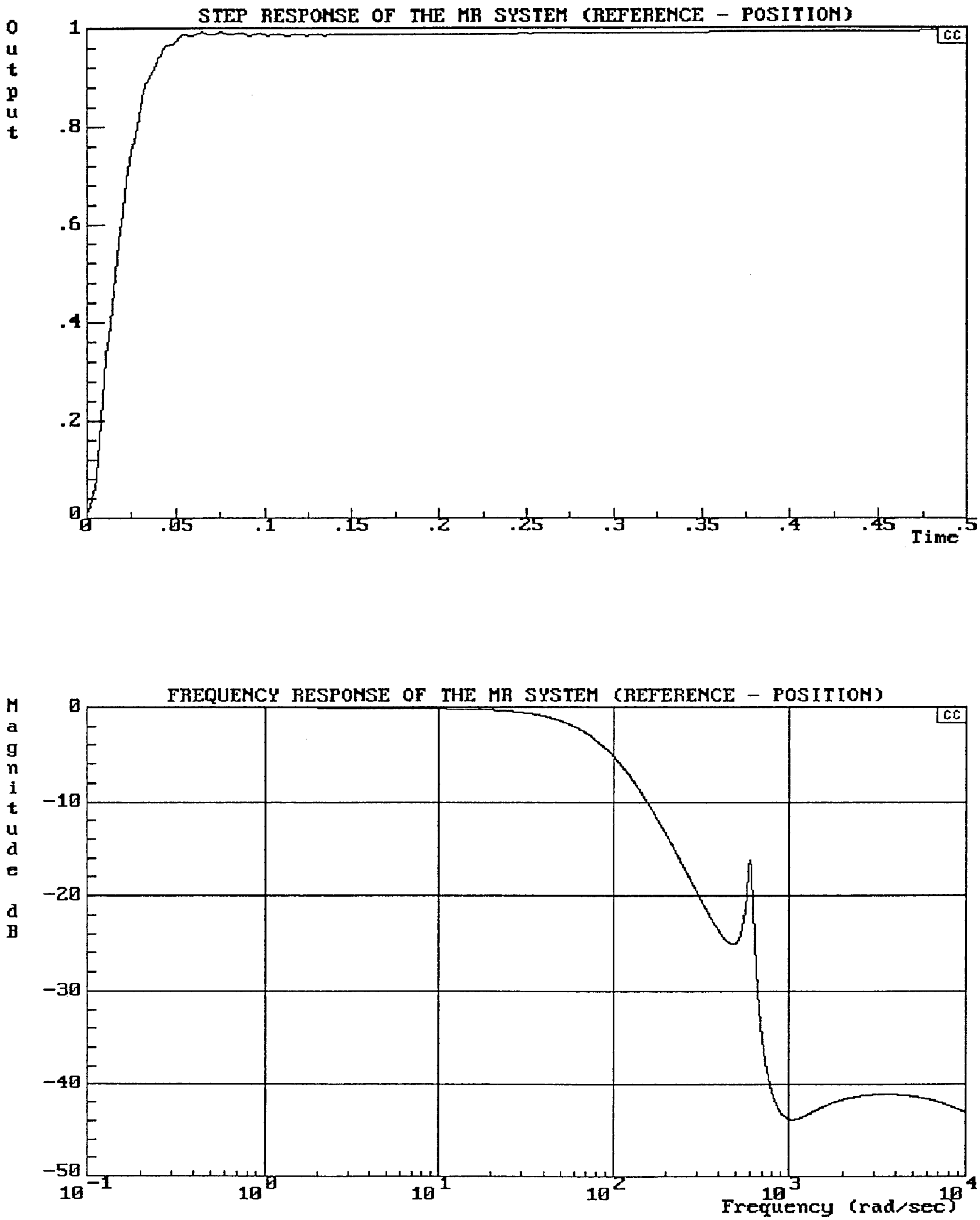
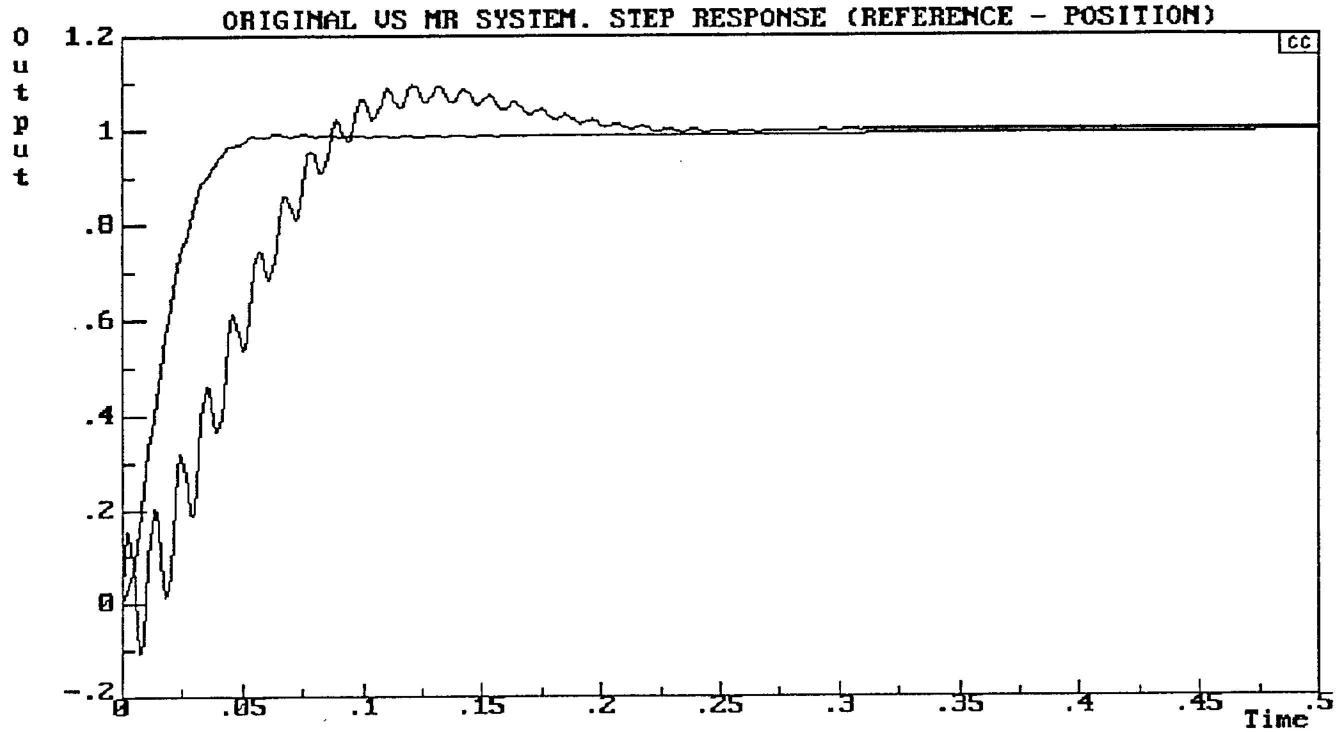

FREQUENCY RESPONSE OF THE REFERENCE MODEL
Magnitude dB
0
-10
-20
-30
-40
-50
-60
-70
-80
-90
Phase
0
-40
-80
-120
-160
-200
-240
-280
-320
-360
10^-1
10^0
10^1
10^2
10^3
10^4
Frequency (rad/sec)

Figure 5-7

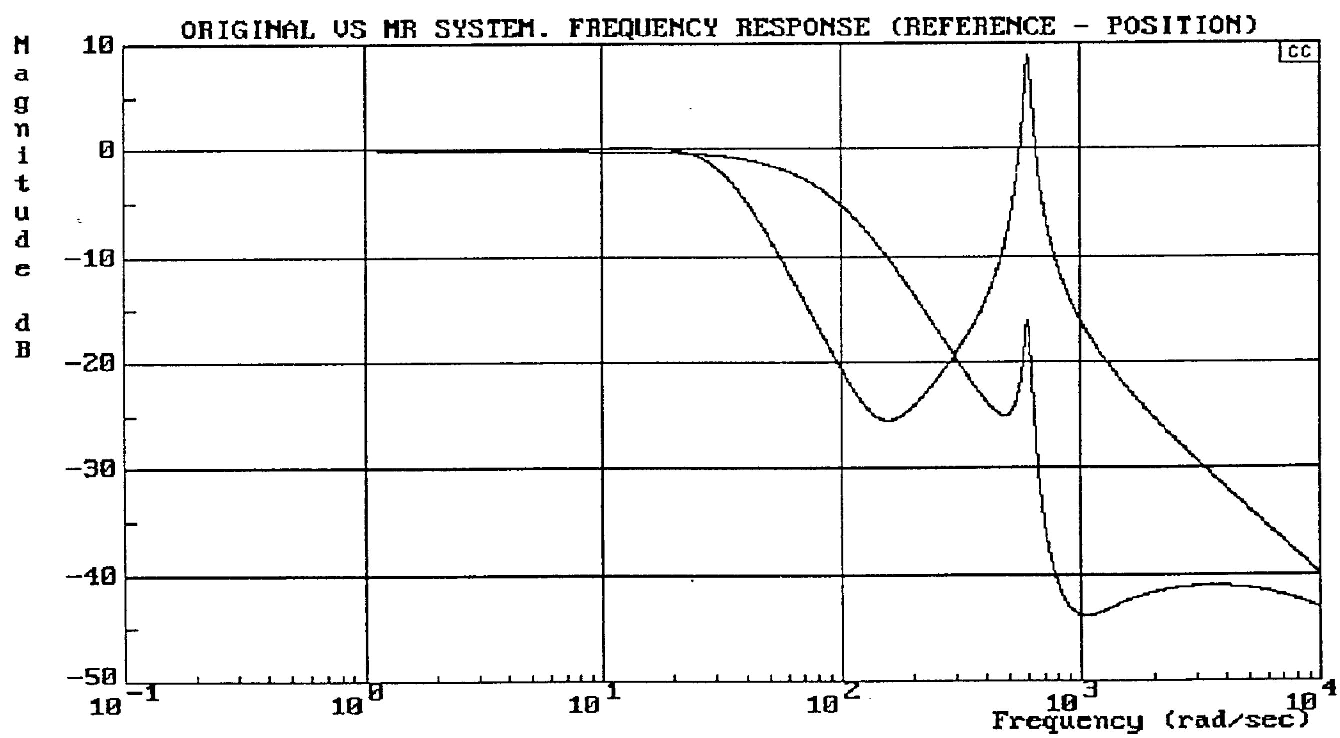


STEP RESPONSE OF THE MR SYSTEM (REFERENCE - POSITION)
Output
1
.8
.6
.4
.2
0
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time
CC
FREQUENCY RESPONSE OF THE MR SYSTEM (REFERENCE - POSITION)
Magnitude dB
0
-10
-20
-30
-40
-50
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)
CC

Figure 5-8



ORIGINAL VS MR SYSTEM. STEP RESPONSE (REFERENCE - POSITION)
Output
1.2
1
.8
.6
.4
.2
0
-.2
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time
CC



ORIGINAL VS MR SYSTEM. FREQUENCY RESPONSE (REFERENCE - POSITION)
Magnitude dB
10
0
-10
-20
-30
-40
-50
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)
CC

Figure 5-9

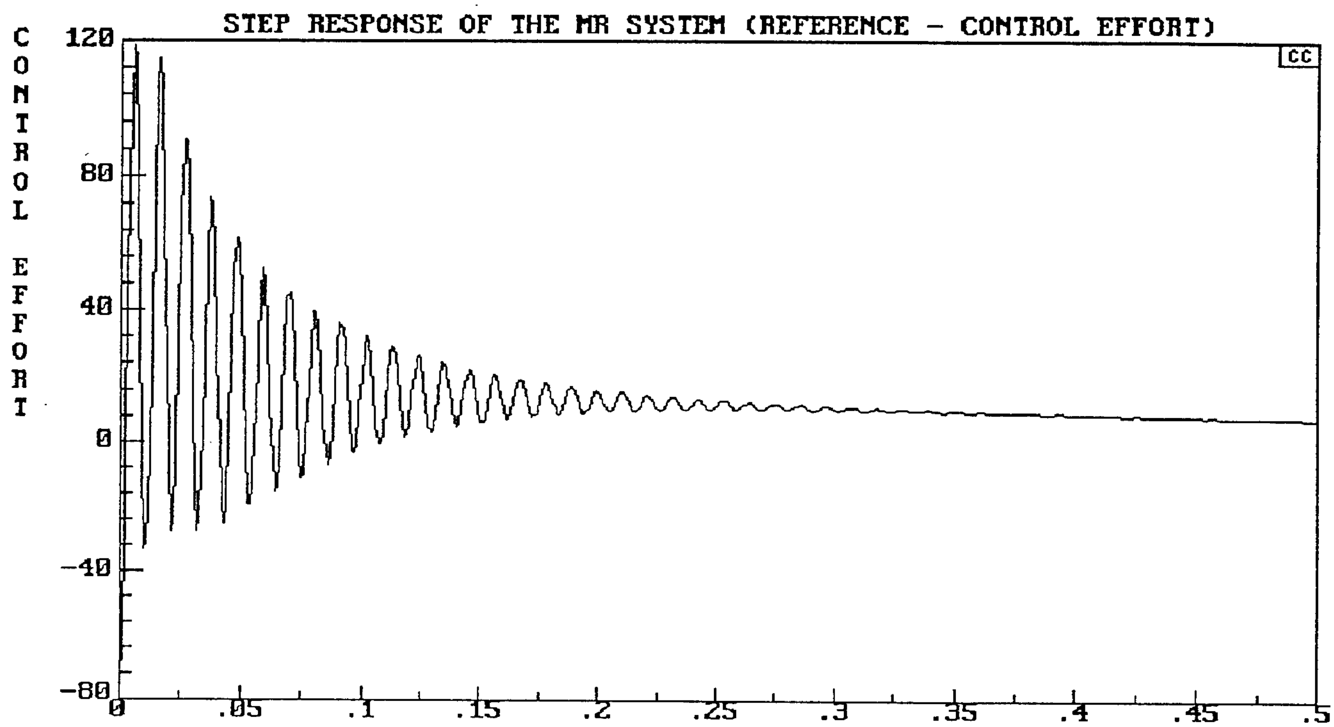
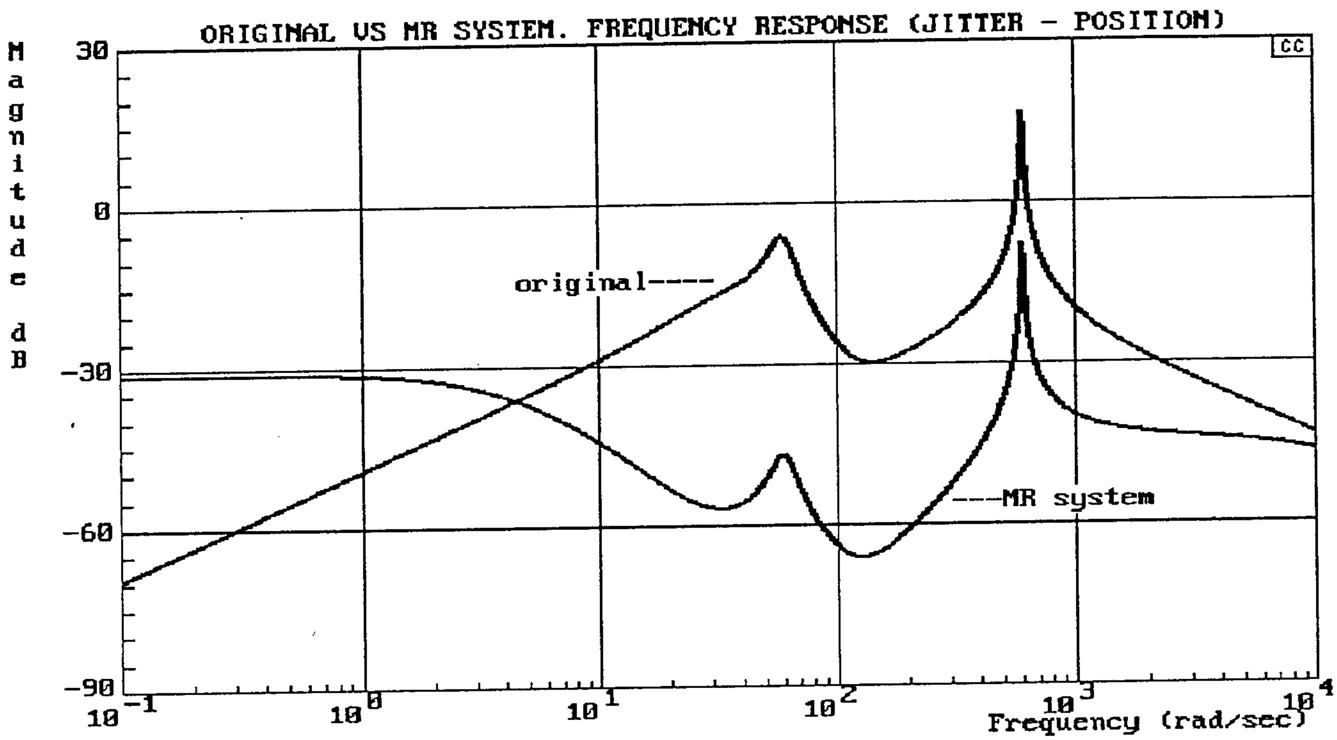
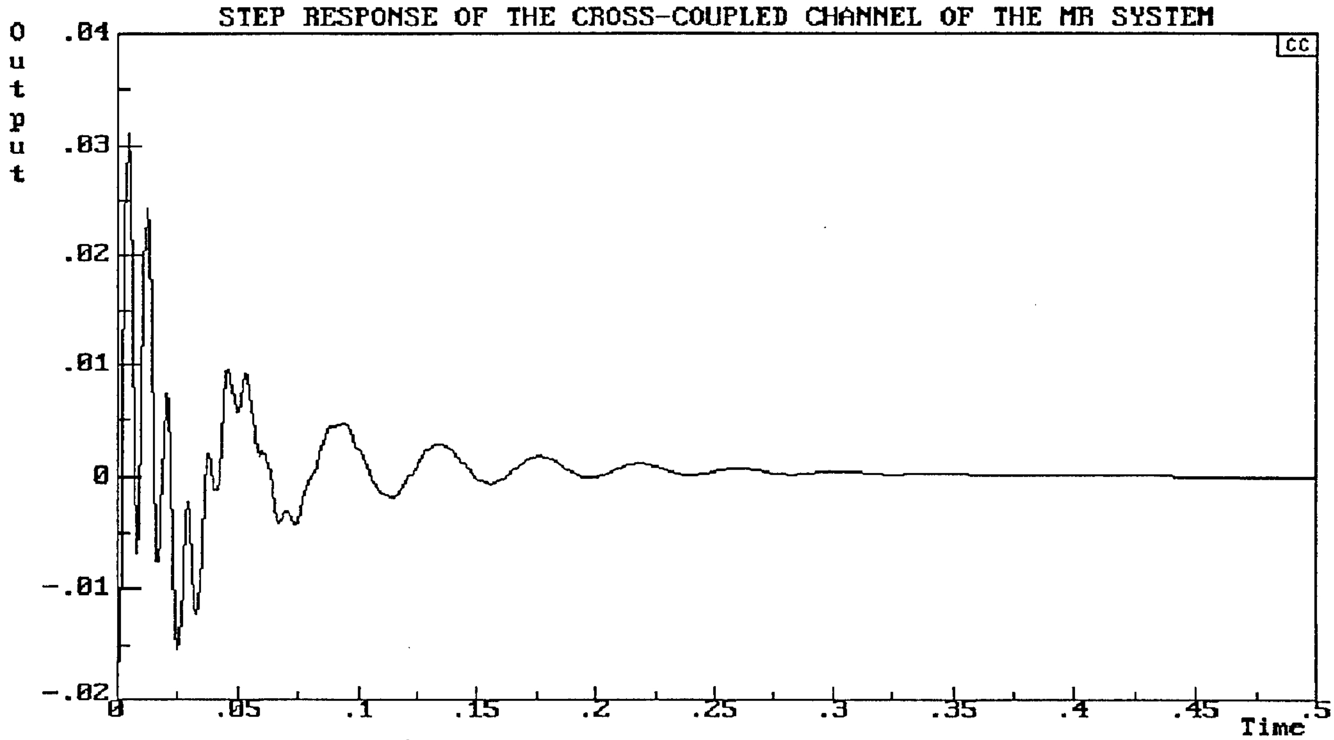

STEP RESPONSE OF THE MR SYSTEM (REFERENCE - CONTROL EFFORT)
CONTROL EFFORT
CC
120
80
40
0
-40
-80
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5

Figure 5-10



ORIGINAL VS MR SYSTEM. FREQUENCY RESPONSE (JITTER - POSITION)
CC
Magnitude dB
30
0
-30
-60
-90
original----
---MR system
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)

Figure 5-11


STEP RESPONSE OF THE CROSS-COUPLED CHANNEL OF THE MR SYSTEM
CC
Output
.04
.03
.02
.01
0
-.01
-.02
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time

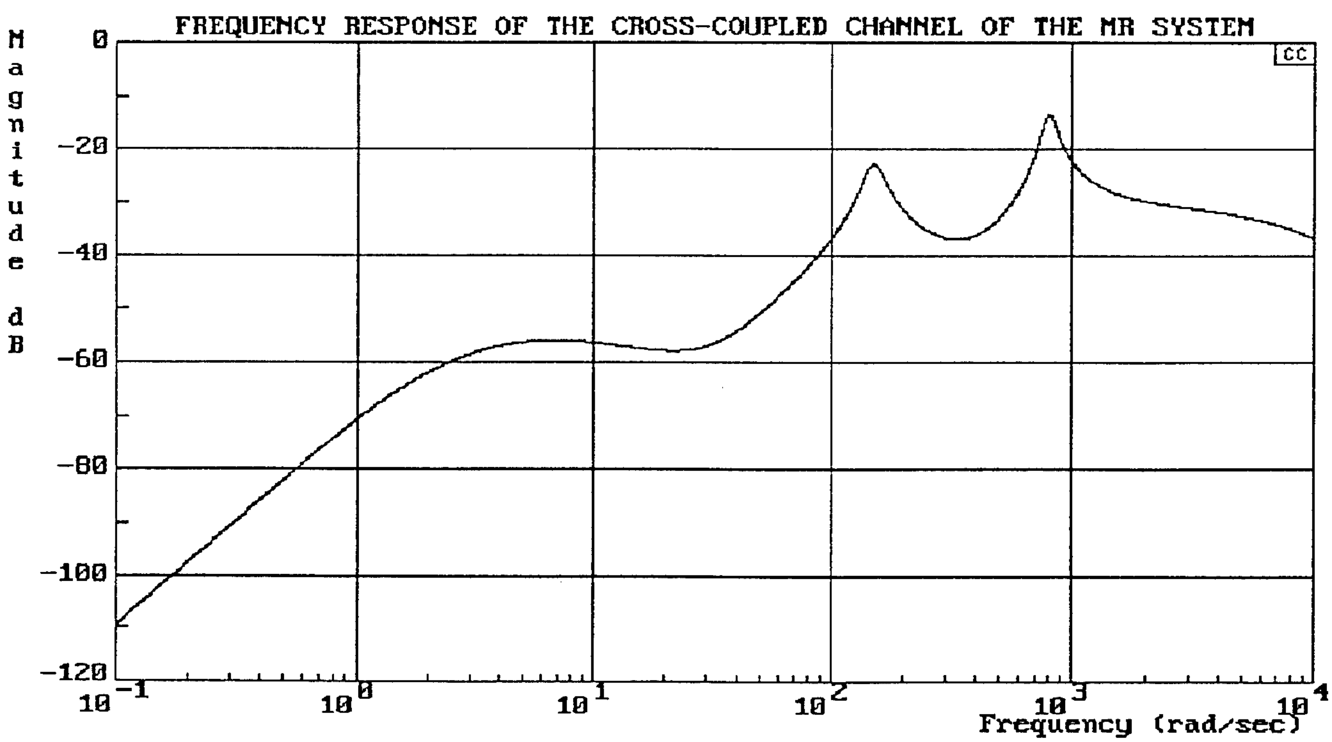
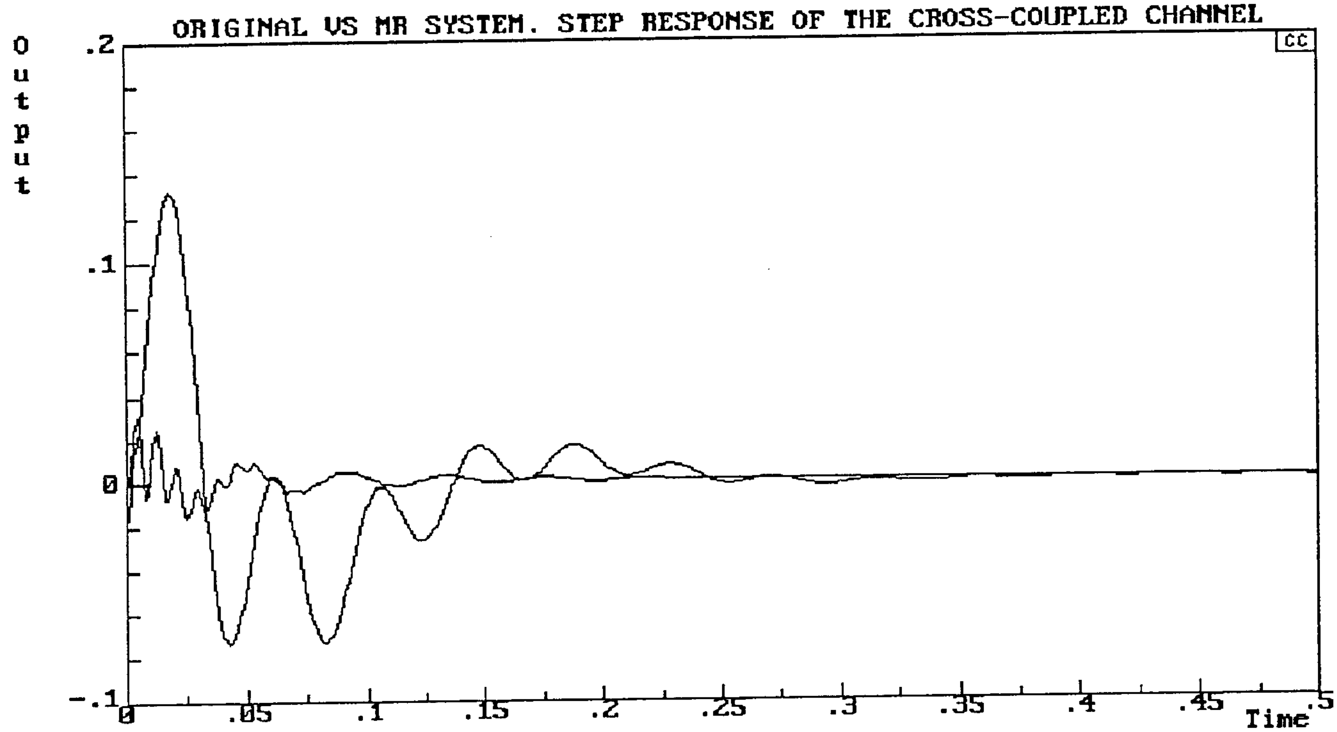
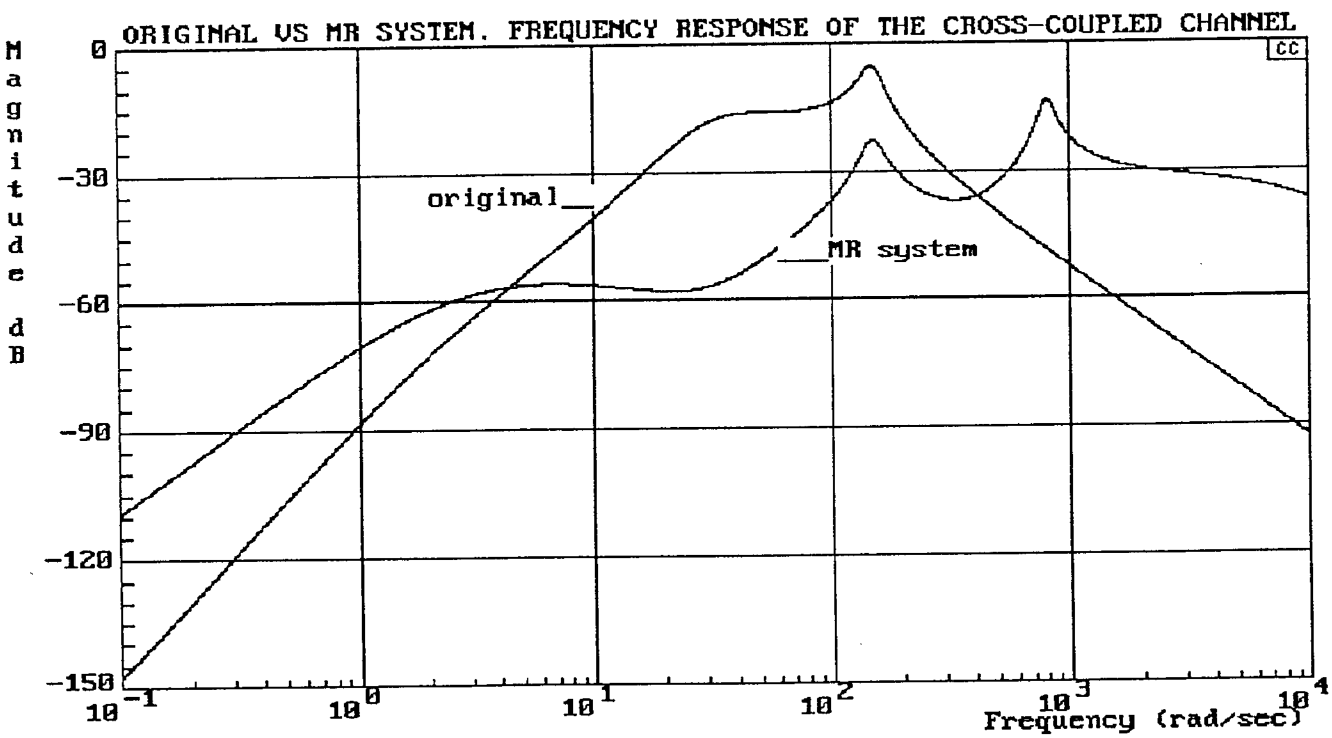

FREQUENCY RESPONSE OF THE CROSS-COUPLED CHANNEL OF THE MR SYSTEM
CC
Magnitude dB
0
-20
-40
-60
-80
-100
-120
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)

Figure 5-12

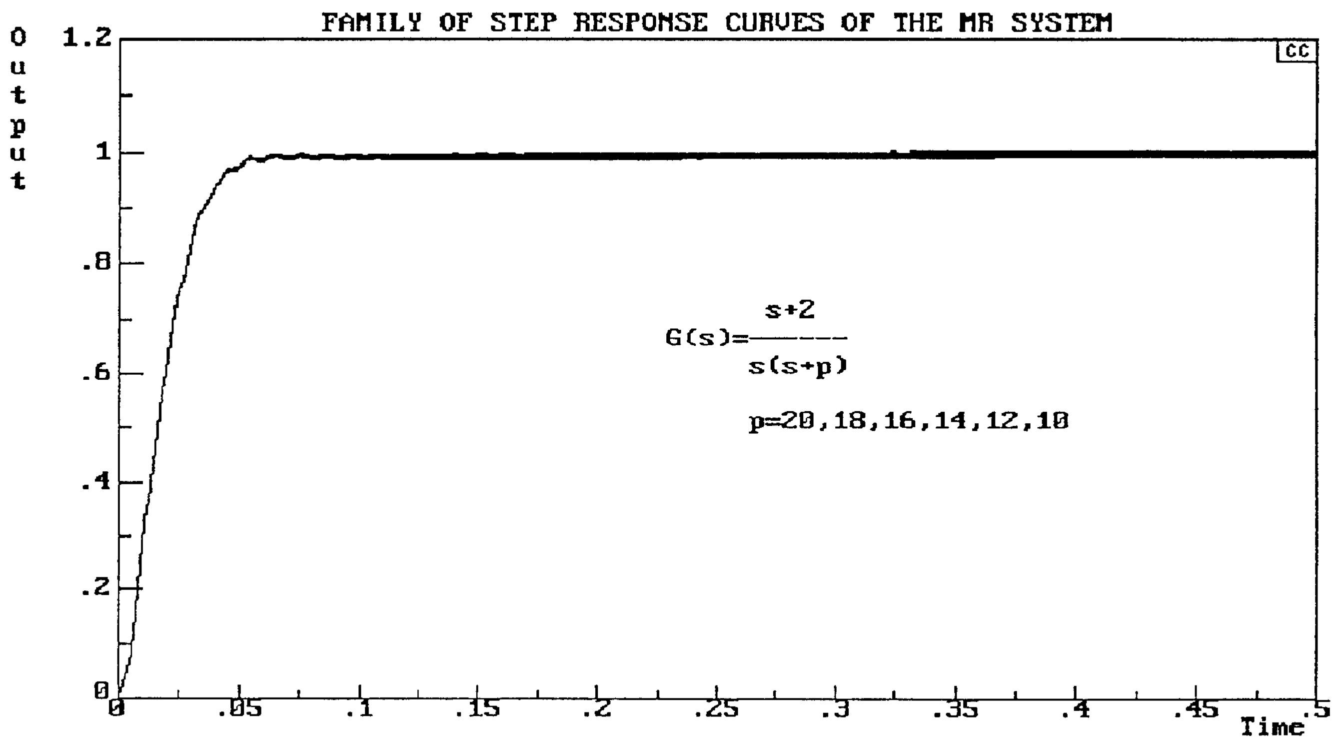


ORIGINAL VS MR SYSTEM. STEP RESPONSE OF THE CROSS-COUPLED CHANNEL
CC
Output
.2
.1
0
-.1
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time



ORIGINAL VS MR SYSTEM. FREQUENCY RESPONSE OF THE CROSS-COUPLED CHANNEL
CC
Magnitude dB
0
-30
-60
-90
-120
-150
original
MR system
10^{-1}
10^{0}
10^{1}
10^{2}
10^{3}
10^{4}
Frequency (rad/sec)

Figure 5-13


FAMILY OF STEP RESPONSE CURVES OF THE MR SYSTEM
CC
Output
1.2
1
.8
.6
.4
.2
0
0
.05
.1
.15
.2
.25
.3
.35
.4
.45
.5
Time
G(s)=s+2 / s(s+p)
p=20,18,16,14,12,10

7. REFERENCES

1. *Skormin, V.A., Tascillo, M.A., Herman, C.R., and Nicholson, D.J.*, "Mathematical Modeling and Simulation Analysis of a Pointing, Acquisition and Tracking System for Laser-Based Intersatellite communication", Optical Engineering, November 93.

Also in the Milestone Series volume "Selected Papers on Precision Stabilization and Tracking Systems for Acquisition, Pointing, and Control Applications"

2. *Skormin, V.A., Tascillo, M.A., and Nicholson, D.J.*, "A Jitter Rejection Technique in a Satellite-Based Laser Communication Systems", Optical Engineering, November 93.

3. *Skormin, V.A., Busch, T.E., and Tascillo, M.A.*, "An Adaptive Jitter Rejection Technique Applicable to Airborne Laser Communication System", Optical Engineering, 1995

Also in the Milestone Series volume "Selected Papers on Precision Stabilization and Tracking Systems for Acquisition, Pointing, and Control Applications"

4. *Skormin V.A., T.E. Busch, and M.A. Givens*, "Model Reference Approach for Compensation of Bending Modes in Fine Steering Mirrors", SPIE Free-Space Laser Communications 1995.

5. *Skormin, V.A., Busch, T.E., Givens M.A.*, "Model Reference Control of a Fast Steering Mirror of a Pointing, Acquisition and Tracking System for Laser Communication", Proceedings of the National Aerospace Electronics Conference NAECON'95, Dayton OH, May 95.

6. *Skormin, V.A., Busch, T.E., and Tascillo, M.A.*, "Demonstration of Jitter Rejection Technique for Free Space Laser Communication", will appear in IEEE Transactions on Aerospace and Electronic Systems, April 1996.

7. *Skormin, V.A.*, "Mathematical Modeling, Simulation, Analysis and Control of a Satellite-Based Laser Communication System", Report to SRI International (Subcontract No. C-b0189/ SRI Project ECU-4007), Binghamton, NY, July 1993

8. *Skormin, V.A.*, "Optical Crosslink Control System Modeling", Final Report to CALSPAN-UB Research Center, Berkshire,

NY, January 1995

9. *Peter M. Thompson*, "Program CC. Version 4", Systems Technology, Inc. Hawthorne, CA

10. *Homayoon Ansari*, "Digital Control Design of a CCD-Based Tracking Loop for Precision Beam Pointing", SPIE Proceedings, Vol. 2123

11. *Donald Russell, Homayoon Ansari, and Chien-C. Chen*, "LaserCom Pointing, Acquisition and Tracking Control Using a CCD-Based Tracker", SPIE Proceedings, Vol. 2123

12. Y.D. Landau, "Adaptive Control. The Model Reference Approach", Marcel Dekker, Inc.

8. APPENDIX

I. The CC code for simulation, analysis and design of the MR control system for 500 Hz mirror.

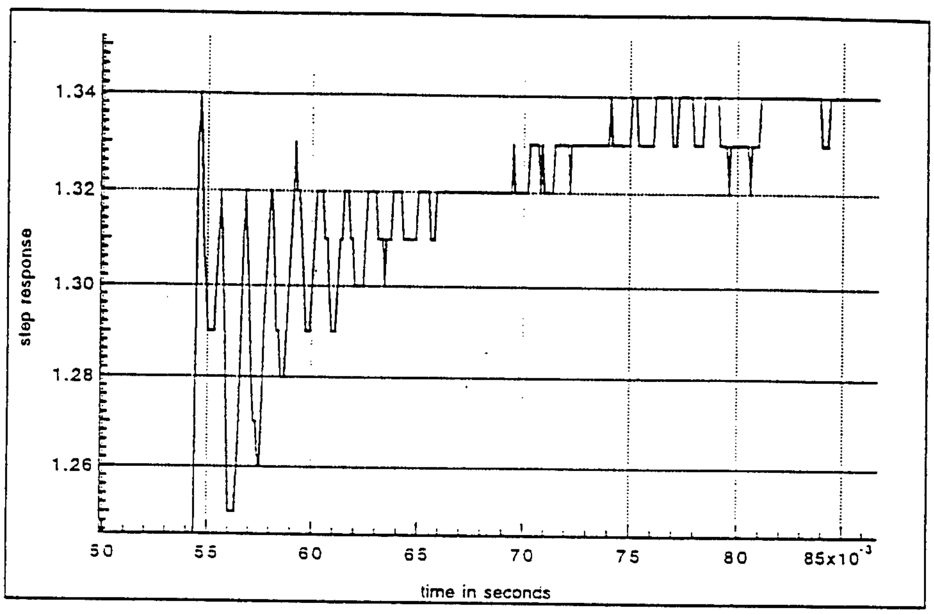
```
echo, Gxx1 - main dynamics
k=.16 & a=200
Gxx1=k*a/(s+a)
echo, Gxx2 - 2-nd order component:
nf=3800   & dr=.5   & k=1
Gxx2=k*nf^2/(s^2+2*nf*dr*s+nf^2)
echo, Gxx3 - ocsillatory terms
fosc=3500 & dr=.03 & k=.000015  & nf=fosc/(1-dr^2)^.5
gxx3=s*k*nf^2/(s^2+2*nf*dr*s+nf^2)
echo, Gxx4 - the drift term
Gxx4=.3/s
Gxx=Gxx1+Gxx2+Gxx3+Gxx4
simul,Gxx,3,.00001,1,0,.05,10,1,a
freq,gxx,.1,10000,10000,2
H=(1+.001*S)/(1+.0001*S)
ECHO,bode,redo
echo,h=2+.02*s
rm=1.17*6000/(s+6000)
g1=(1+rm*h)*(Gxx|h)
dum=Gxx*h
g2=(1+rm*h)*(1|dum)
e=rm-gxx
simul,g1,3,.00001,1,0,.005,10,1,a
```

II. The CC code for simulation, analysis and design of the high precision MR position control system .

```
echo, g - transfer function describing main dynamics (channel 1)
g=(s+2)/(s+20)/s
gv1=(s+2)/(s+18)/s
gv2=(s+2)/(s+16)/s
gv3=(s+2)/(s+14)/s
gv4=(s+2)/(s+12)/s
gv5=(s+2)/(s+10)/s
echo, b,b1 - transfer functions describing bending modes
echo, in two adjacent dynamic channels
b=100*s/(s^2+2*.03*600*s+600^2)
b1=126*s/(s^2+2*.07*800*s+800^2)
echo, gj - transfer function describing jitter effects
gj=310/(s^2+2*.103*60*s+60^2)+66*s/(s^2+2*.008*600*s+600^2)
echo, gc - transfer function describing cross-coupling
gc=2.3*s/(s^2+2*.1*150*s+150^2)
echo, h - transfer function of the forward path filter
h=(s+20)/(s+40)
echo, f - transfer function of the feedback controller
f=1000/(s+2)
echo, p - transfer function of the filter in the reference
echo, channel
p=1000/(s+2)
echo, m - transfer function of the reference model
m=100^2/(s^2+2*.9*100*s+100^2)
echo, q - transfer function of the adaptation mechanism
q=1000*(1+.1*s)/(1+.01*s)
g0=(g*h)|f
g01=(gv1*h)|f
g02=(gv2*h)|f
g03=(gv3*h)|f
g04=(gv4*h)|f
g05=(gv5*h)|f
```

```
echo, transfer function of the existing system, ref-position:
ge=p*g0+b
gev1=p*g01+b
gev2=p*g02+b
gev3=p*g03+b
gev4=p*g04+b
gev5=p*g05+b
echo, transfer function of the existing system, ref-control
echo, effort:
geu=p*h|(f*g)
echo, transfer function of the existing system, jitt-position:
gejit=gj*(1|(h*g*f))
echo, transfer function of the MR system, cross-coupling effort:
gecc=geu*gc*(1|(h*g*f))
echo, transfer function of the MR system, ref-position:
gmr=(p*g0+m*q*g0+b)/(1+g0*q)
gmr1=(p*g01+m*q*g01+b)/(1+g01*q)
gmr2=(p*g02+m*q*g02+b)/(1+g02*q)
gmr3=(p*g03+m*q*g03+b)/(1+g03*q)
gmr4=(p*g04+m*q*g04+b)/(1+g04*q)
gmr5=(p*g05+m*q*g05+b)/(1+g05*q)
echo, transfer function of the MR system, ref-control effort:
g00=h|(f*g)
gu=g00*(p+m*q-b*q)/(1+g0*q)
gu1=g01*(p+m*q-b1*q)/(1+g0*q)
echo, transfer function of the MR system, jitt-position:
gjit=gj/(1+g0*q)
echo, transfer function of the MR system, cross-coupling effort:
gcc=gu1*gc/(1+g0*q)
echo, simulation of a step response:
simul,gecc,3,.001,1,0,.5,10,1,a
echo, obtaining frequency response and Bode plot:
freq,gecc,.1,10000,5000
bode,3,a
```

Figure A-1. Recorded step response of the azimuth channel



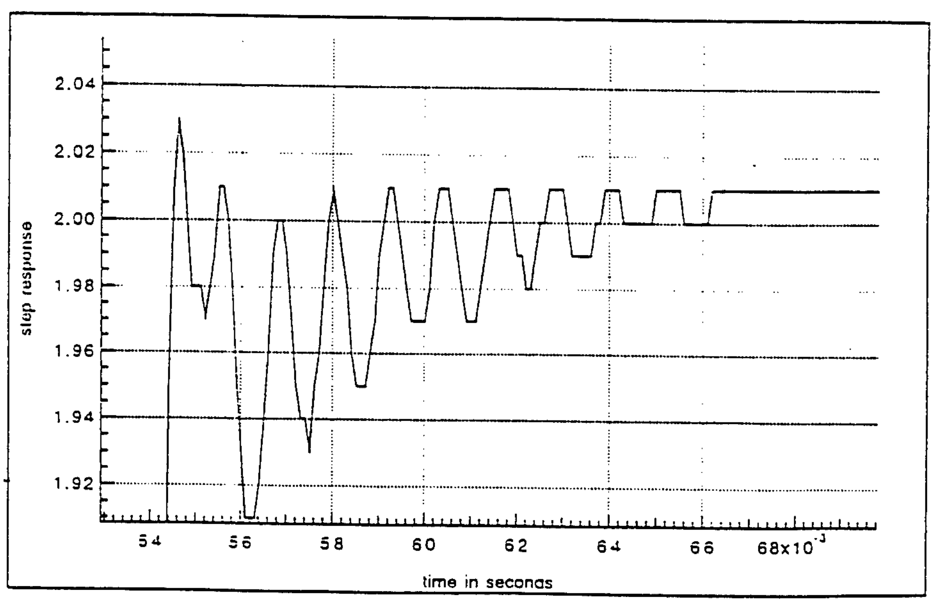
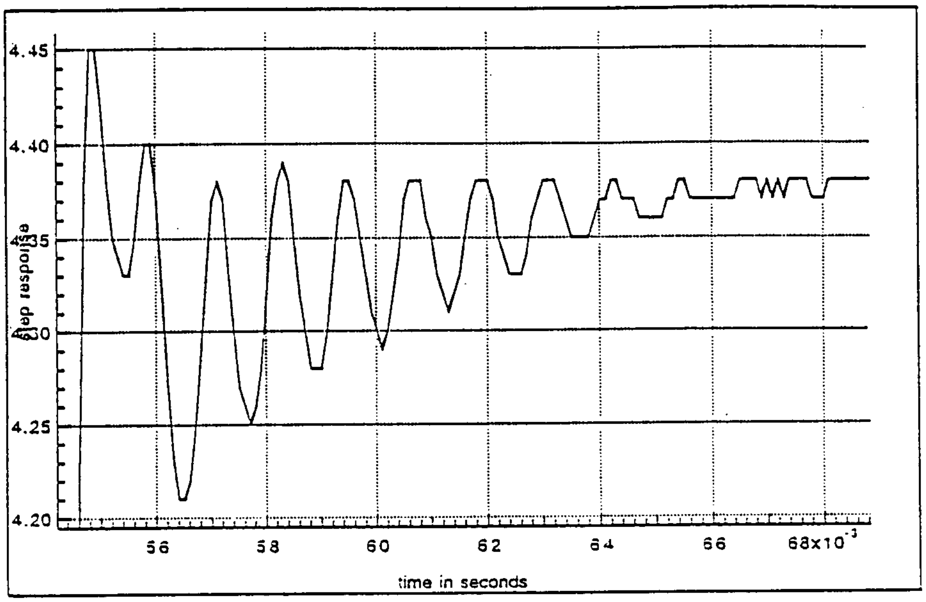


Figure A-2. Recorded step response of the azimuth channel



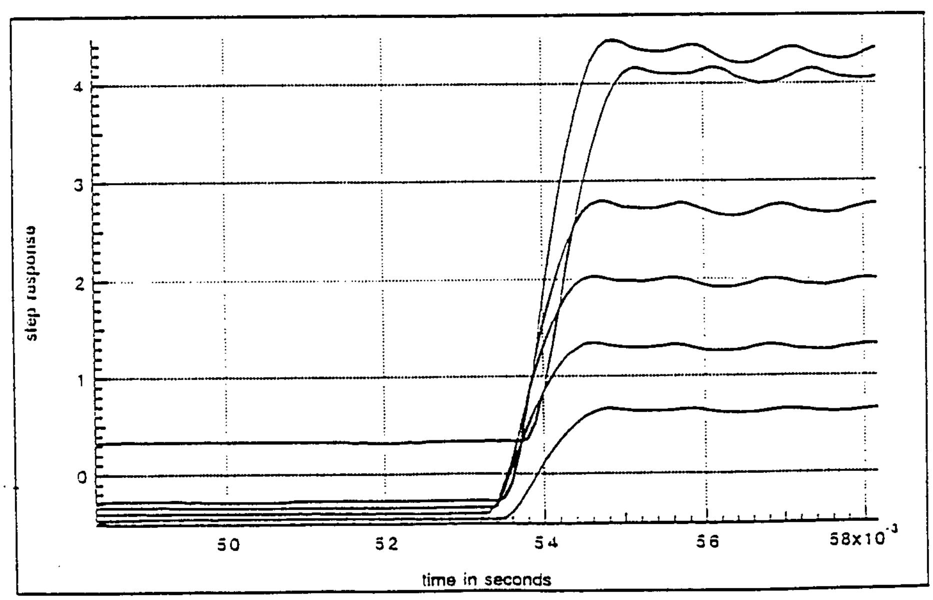
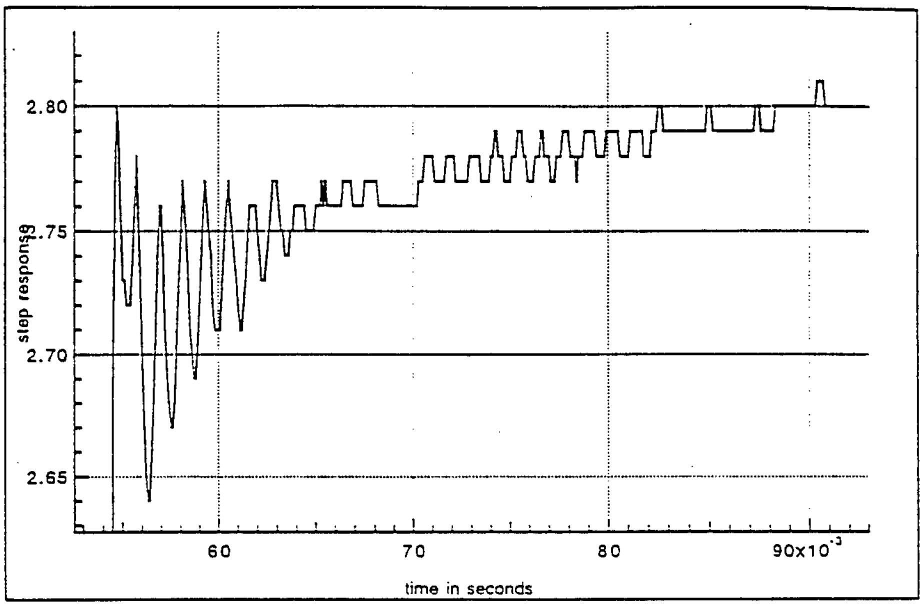


Figure A-3. Recorded step response of the azimuth channel



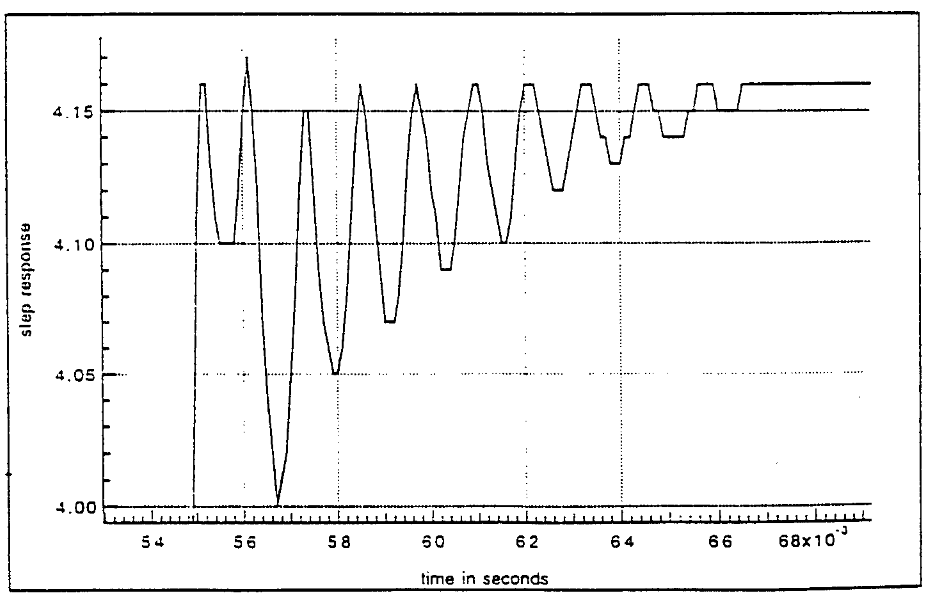
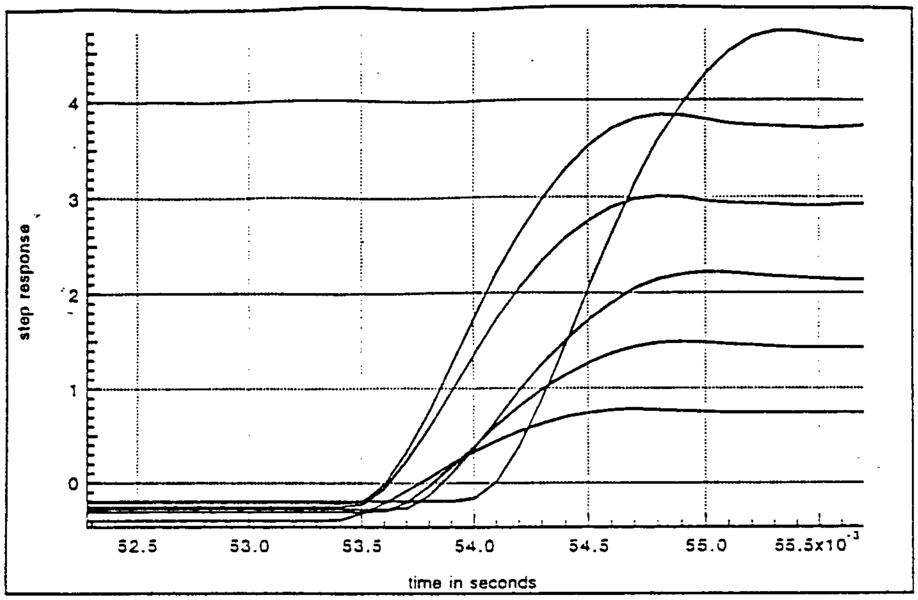


Figure A-4. Recorded step response of the elevation channel



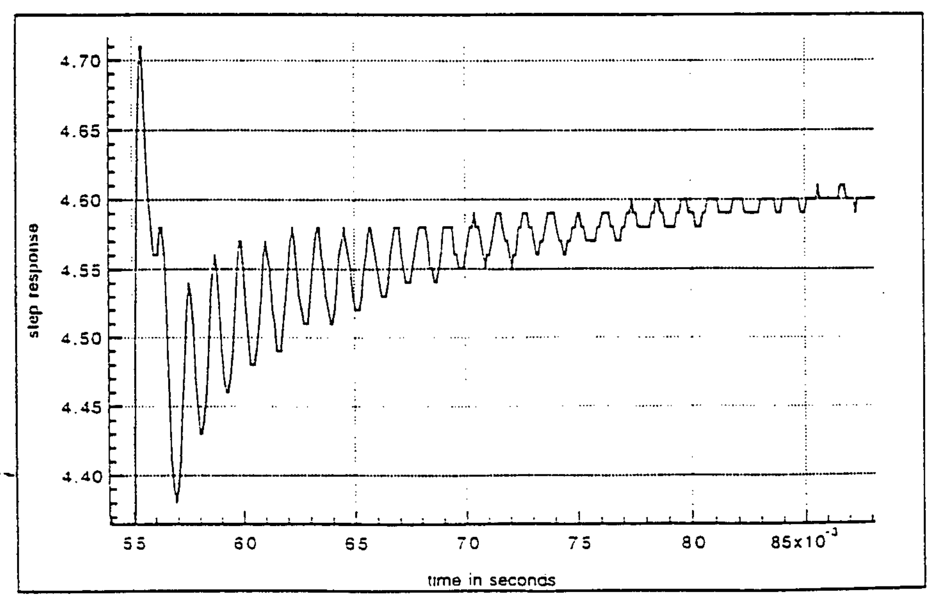
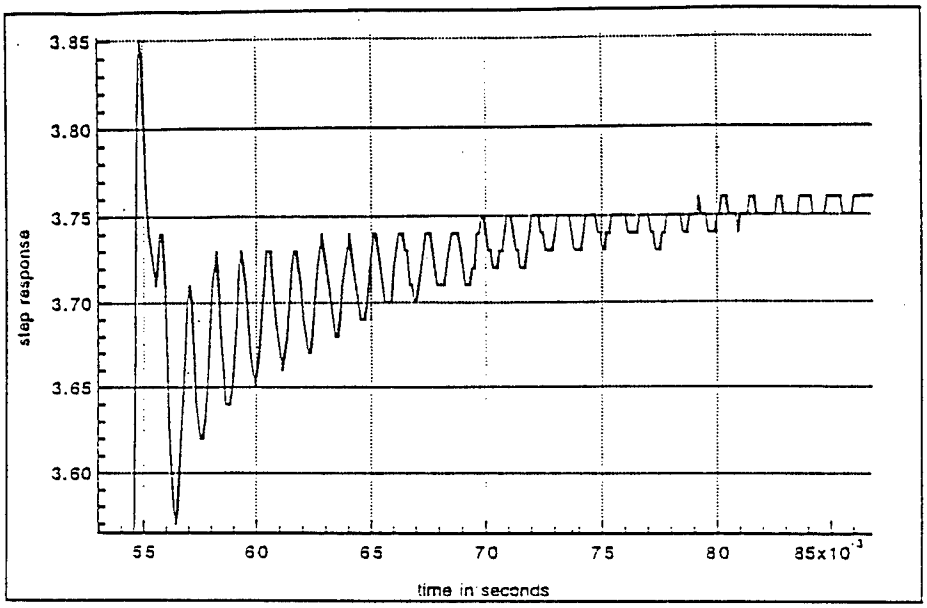


Figure A-5. Recorded step response of the elevation channel



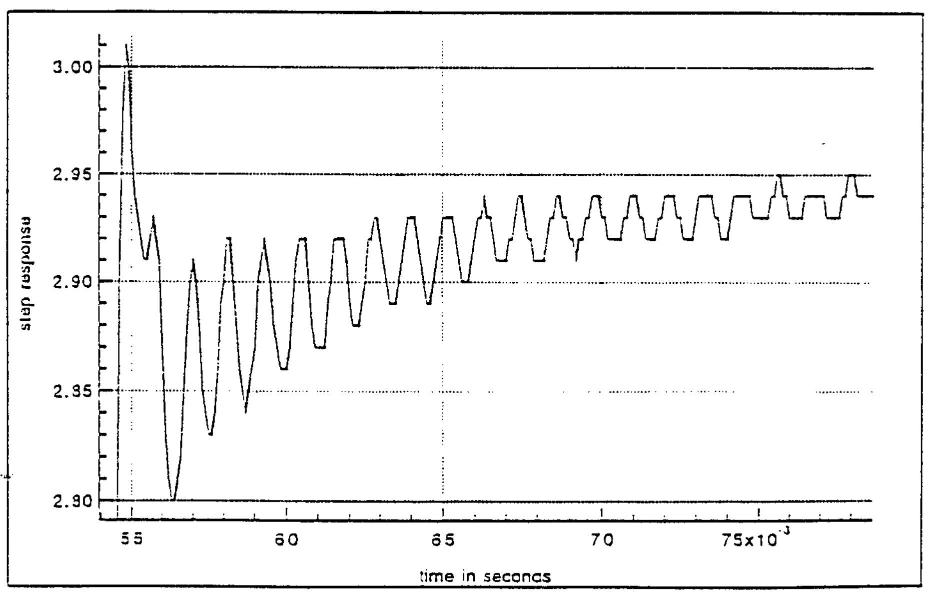
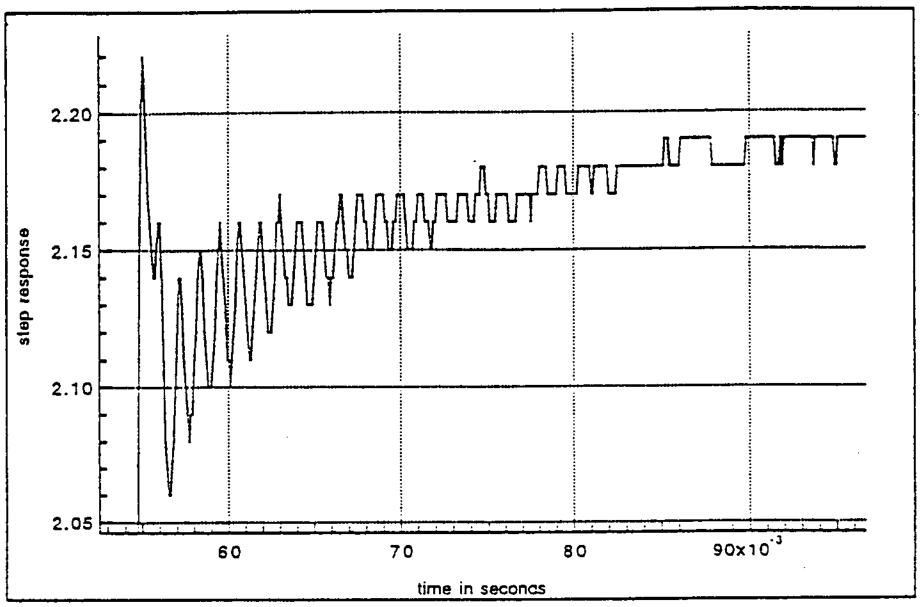


Figure A-6. Recorded step response of the elevation channel



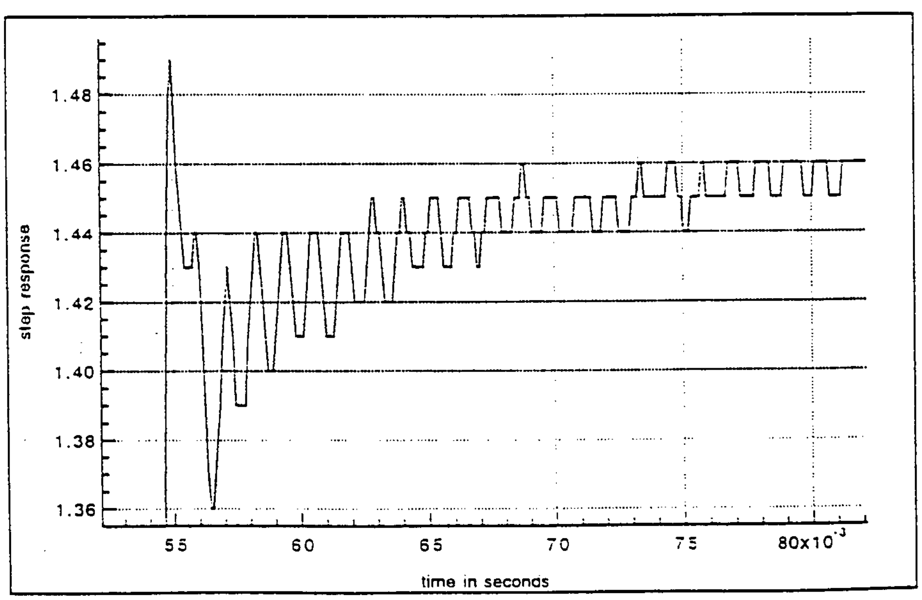


Figure A-7. Recorded interactions between coupled channels

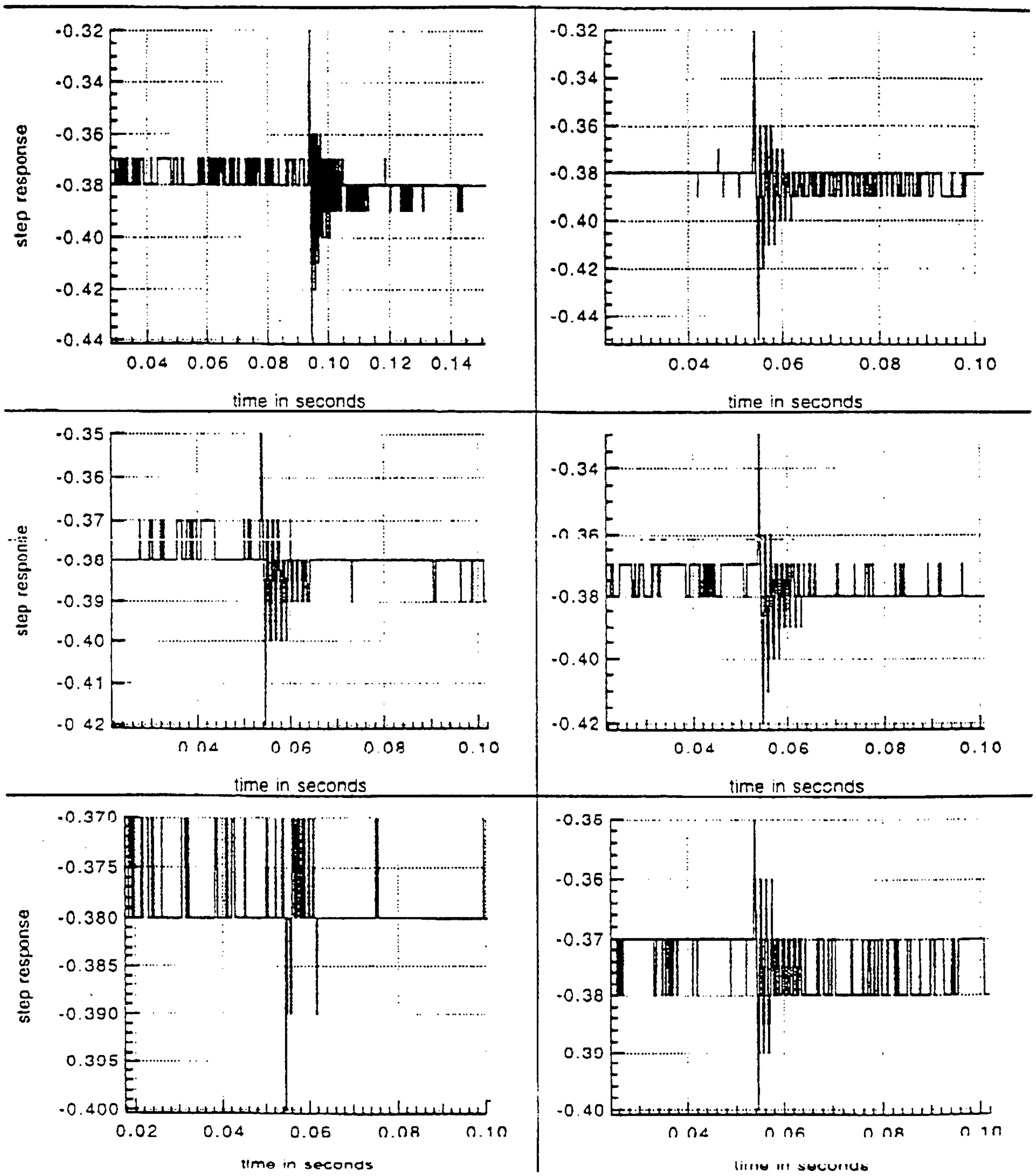


Figure A-8. Recorded interactions between coupled channels

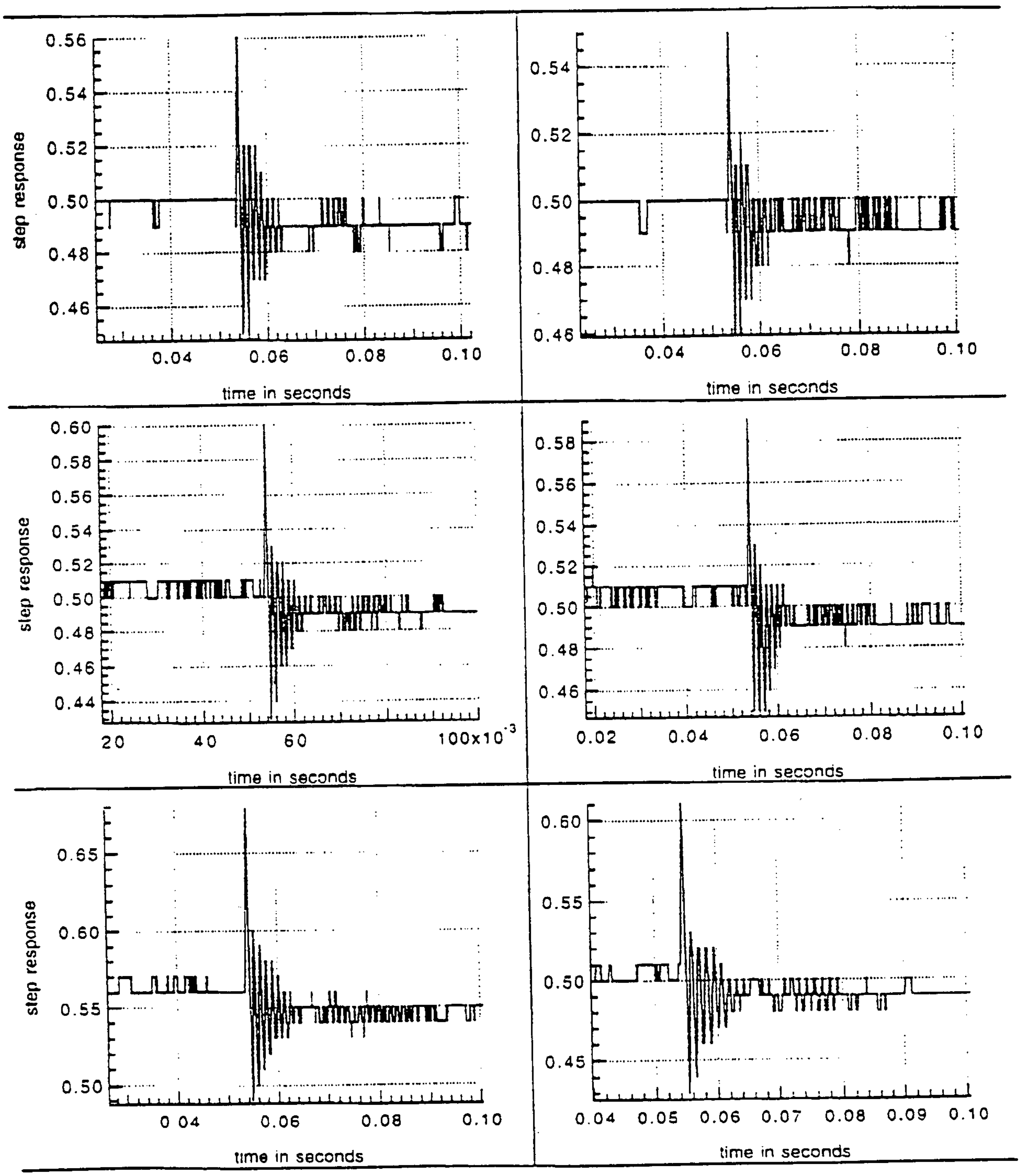


Figure A-9. Recorded interactions between coupled channels

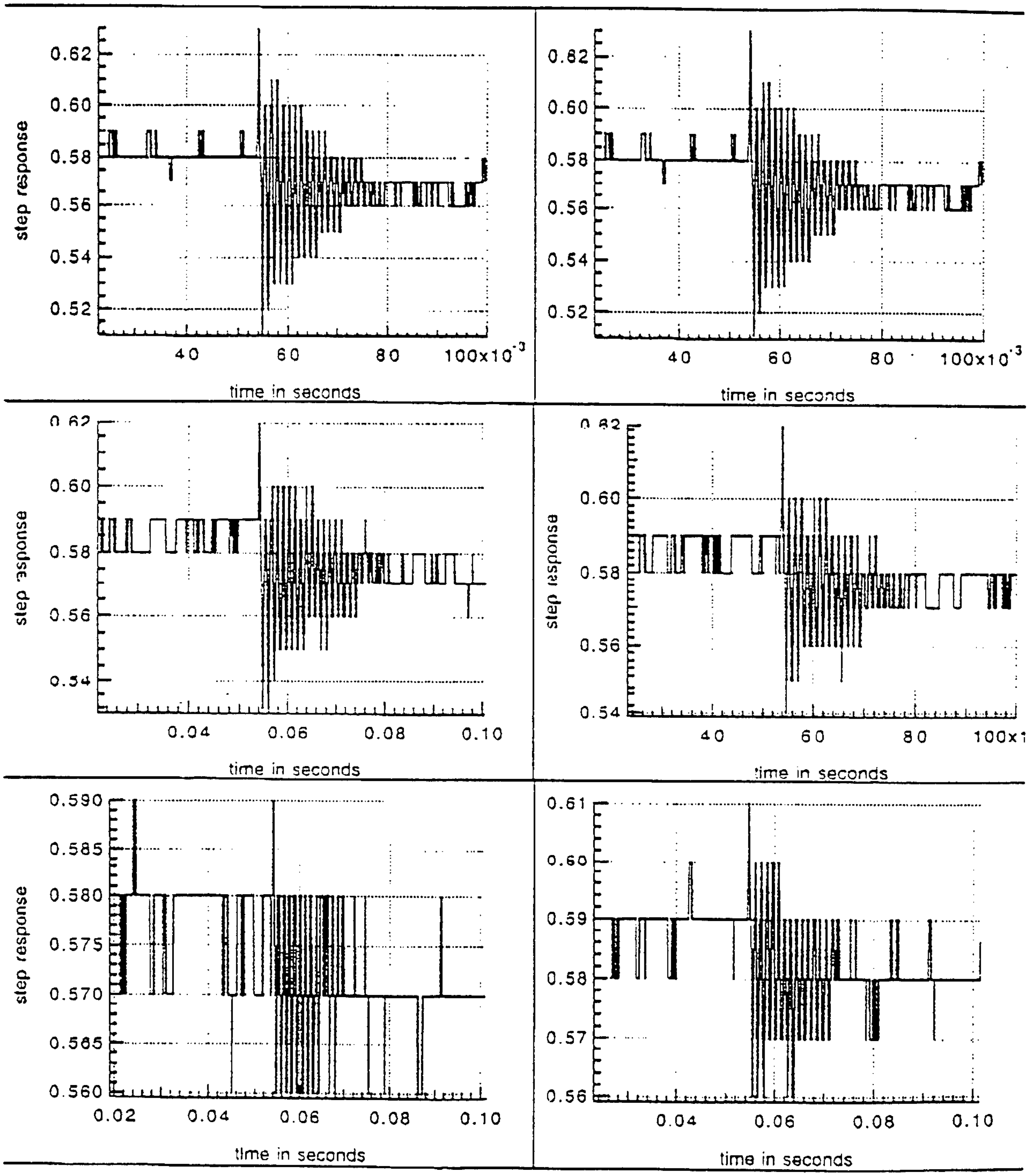


Figure A-10. Derivation of transfer functions of the MR position control system

Equivalent signal-flow graphs of the MR system
The jitter input

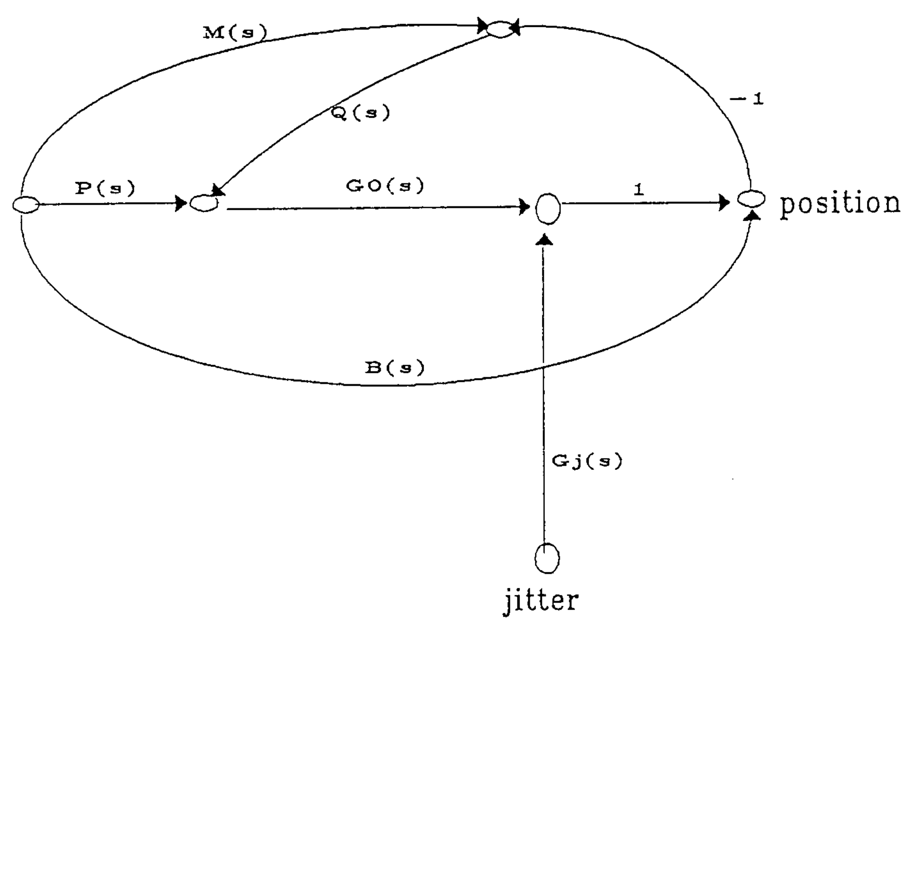


Figure A-11. Derivation of transfer functions of the MR position control system

Equivalent signal-flow graphs of the MR system
The coupling effect

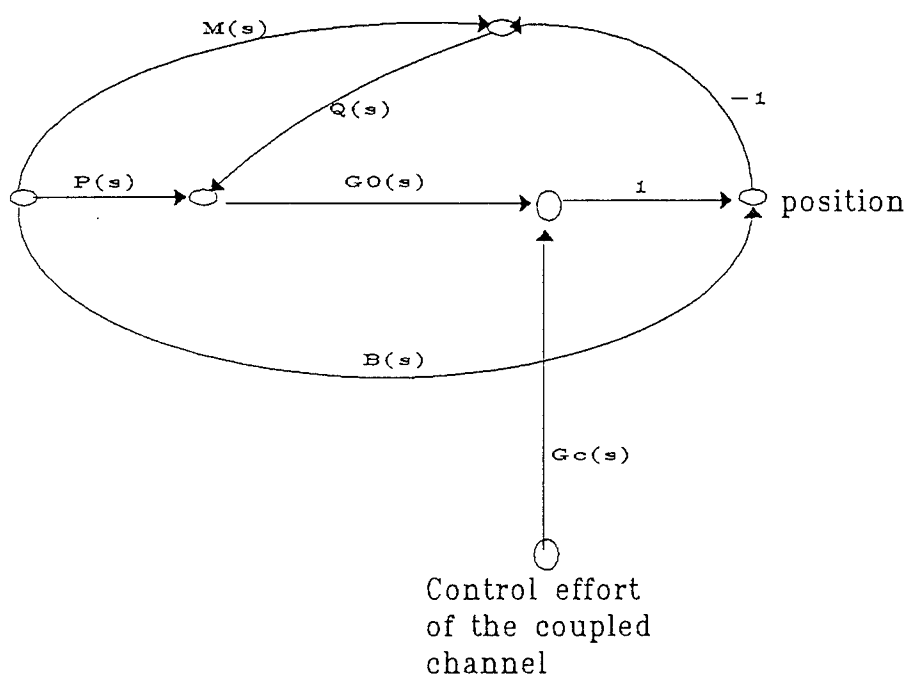


Figure A-12. Derivation of transfer functions of the MR position control system

Equivalent signal-flow graphs of the MR system
Definition of the control effort

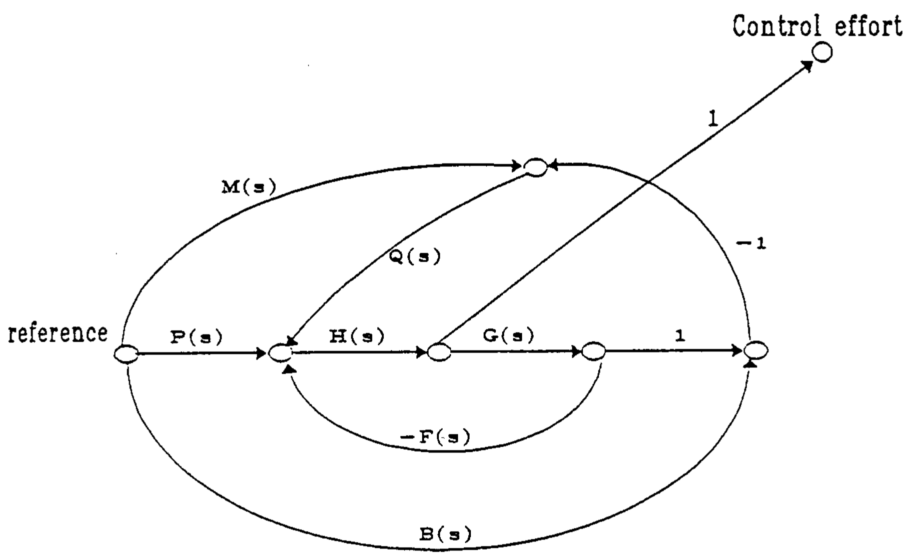
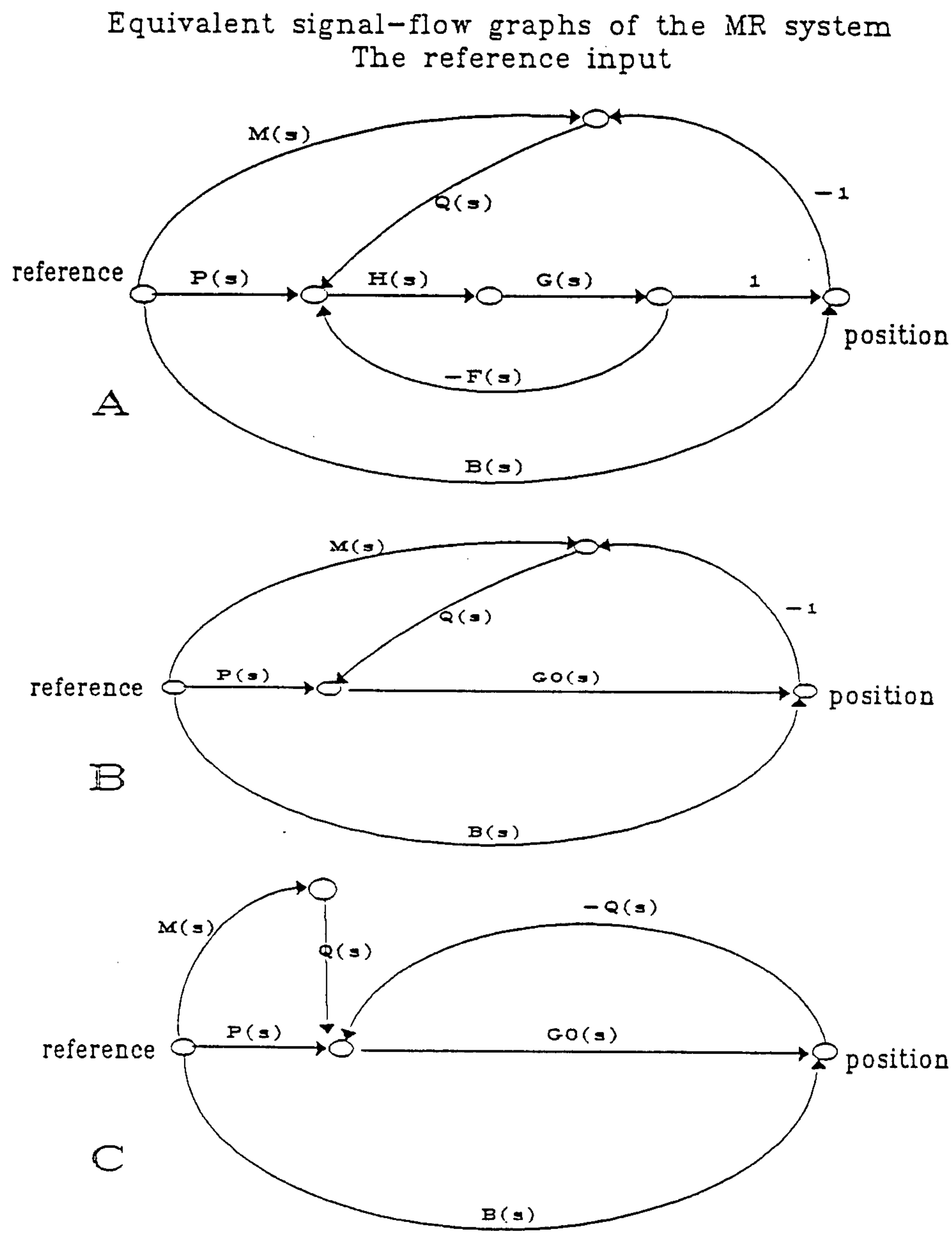


Figure A-13. Derivation of transfer functions of the MR position control system



Derivation of transfer functions of the MR position control system using the signal-flow graphs

The reference-to-position transfer function:

Path #1: $P(s)*G_0(s)$
Path #2: $M(s)*Q(s)*G_0(s)$
Path #3: $B(s)$
Loop #1: $-G_0(s)*Q(s)$

$$G_{MR}(s) = \frac{P(s)*G_0(s)+M(s)*Q(s)*G_0(s)+B(s)}{1+G_0(s)*Q(s)}$$

where $G_0(s) = \dfrac{H(s)*G(s)}{1+H(s)*G(s)*F(s)}$

Let us check on the original signal-flow graph:

Path #1: $P(s)*H(s)*G(s)$
Path #2: $M(s)*Q(s)*H(s)*G(s)$
Path #3: $B(s)$
Loop #1: $-H(s)*G(s)*F(s)$
Loop #2: $-H(s)*G(s)*Q(s)$

$$G_{MR}(s) = \frac{P(s)*H(s)*G(s)+M(s)*Q(s)*H(s)*G(s)+B(s)\,[1+H(s)*G(s)*F(s)]}{1+H(s)*G(s)*F(s)+H(s)*G(s)*Q(s)}$$

$$= \frac{P(s)\dfrac{H(s)*G(s)}{1+H(s)*G(s)*F(s)} + M(s)Q(s)\dfrac{H(s)*G(s)}{1+H(s)*G(s)*F(s)} + B(s)}{\dfrac{1+H(s)*G(s)*F(s)}{1+H(s)*G(s)*F(s)} + Q(s)\dfrac{H(s)*G(s)}{1+H(s)*G(s)*F(s)}}$$

Definition of the control effort:

Path #1: $P(s)*H(s)$
Path #2: $M(s)*Q(s)*H(s)$
Path #3: $-B(s)*Q(s)*H(s)$
Loop #1: $-H(s)*G(s)*F(s)$
Loop #2: $-H(s)*G(s)*Q(s)$

$$G_U(s) = \frac{P(s)*H(s)+M(s)*Q(s)*H(s)-B(s)*Q(s)*H(s)}{1+H(s)*G(s)*F(s)+H(s)*G(s)*Q(s)}$$

introduce $G_1(s) = \dfrac{H(s)}{1+H(s)*G(s)*F(s)}$, then

Note that $G_1(s)G(s)=G_0(s)$, therefore,

$$G_U(s)=G_1(s)\,\frac{P(s)+M(s)*Q(s)-B(s)*Q(s)}{1+Q(s)*G_0(s)}$$

Definition of the jitter effects:

Path #1: $G_j(s)$
Loop #1: $-G_0(s)*Q(s)$

$$G_{JIT}(s) = \frac{G_j(s)}{1+G_0(s)*Q(s)}$$

Analysis of the cross-coupling effect:

Path #1: $G_c(s)$
Loop #1: $-G_0(s)*Q(s)$

$$G_{CC}(s)=G_U(s)\,\frac{G_c(s)}{1+G_0(s)*Q(s)}$$

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